STEAM

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PREFACE

THE following pages are based on the notes of lectures recently given by the author to an evening class of young mechanical engineers on steam, steam engines, and boilers.

The rapid progress made in engineering science during recent years, and the limited space at disposal, have necessitated the omission of descriptions of obsolete types of steam engines, and the substitution of other matters of more importance to the present generation of engineering students.

Special prominence has been given to the principles involved in the economical use of steam, and it is hoped that the book may be found of value, not only to the student, but to the practical engineer whose time and opportunities for the study of principles are limited.

Most of the diagrams have been prepared specially for this work. I am, however, indebted to Mr. GEO. C. V. HOLMES for the use of a few diagrams from his work on the Steam Engine.

W. R.

SHEFFIELD : *October* 1889.

CONTENTS



CHAPTER I

PAGE

Introduction—Heat, its nature and effects—Temperature—Thermometers—Specific heat—Absolute temperatures 1

CHAPTER II

Unit of heat and unit of work—Horse-power—Mechanical equivalent of heat 7

CHAPTER III

Transfer of heat—Radiation—Conduction—Convection 12

CHAPTER IV

Combustion of fuel—Heat of combustion—Evaporative power of fuel . . 15

CHAPTER V

Application of heat to solids—Application of heat to gases—Pressure of the air—Absolute pressure—Application of heat to water—Boiling—Condensation of steam—Vacuum 20

CHAPTER VI

Action of heat in the formation of steam—Work done by steam during formation at low- and high-pressures respectively—Efficiency of the steam—Heat rejected by steam to condenser—Sensible heat—Latent heat—Total heat of evaporation . . 28

CHAPTER VII

PAGE

Saturated steam—Table of properties—Water heated in a closed vessel—Temperature of mixtures--Condensing water . . 38

CHAPTER VIII

Relation between the pressure and volume of gases—The hyperbolic curve 42

CHAPTER IX

Expansive working--Work done by steam used expansively—Back pressure—Mean pressure—Indicated horse-power—Examples illustrating economy of expansive working—Limit of useful expansion—Clearance in the cylinder—Priming—Cylinder condensation 48

CHAPTER X

The steam engine--Non-condensing engines—Engine details. The cylinder—Cylinder liner—Steam jacket—Escape valve—Relief cocks. Pistons—Piston speed—Piston displacement—Piston rods—Crossheads and guide blocks—The connecting rod—Relative positions of piston and crank pin—Rotary engines . 69

CHAPTER XI

The slide valve—Lap, lead, angular advance—Piston valves—The double-ported slide valve—To set a slide valve—Eccentrics—Reversing gear—The link motion 89

CHAPTER XII

Cranks and crank shafts—Tangential pressure on crank pin—Shaft couplings—Journals—Bearings 101

CHAPTER XIII

Condensers—The jet condenser—The air-pump—The surface condenser—The vacuum gauge—Pumps 109

CHAPTER XIV

PAGE

Governors—The Watt governor—The Porter governor with automatic expansion gear—Fly wheels—The locomotive engine, arrangement and construction of 119

CHAPTER XV

Compound engines—compared with single cylinder engine—The two-cylinder compound engine illustrated 129

CHAPTER XVI

Types of compound engines—The Woolf engine—Distribution of steam—The range of temperature—The distribution of stresses—The Receiver engine—Distribution of the steam—Triple and Quadruple expansion engines. Economy due to compounding 139

CHAPTER XVII

Boilers—Resistance of cylindrical vessels—Descriptions of boilers—The Cornish boiler—Water tubes—The Lancashire boiler—The vertical boilers—Marine boilers—Steam room—The locomotive boiler—Roof stays—Bar stays—Heating surface of tubes—Safety valves—To graduate the lever—Spring loaded valves—The dead-weight safety valve—Bourdon's pressure gauge—Boiler performance and efficiency 158

CHAPTER XVIII

Practical notes on the care and management of engines and boilers—Annual inspection of engines and boilers 181

APPENDIX:—QUESTIONS AND EXERCISES 186

INDEX 200

STEAM



CHAPTER I

INTRODUCTION

THE object of the study of steam and its applications is to obtain from the steam engine the greatest possible amount of work for the least possible expenditure of fuel.

In order to understand the principles which underlie the economical production and use of steam, we shall consider the following subjects, and in the order given, viz. :

1. Heat.
2. Steam.
3. Engines.
4. Boilers.

HEAT, ITS NATURE AND EFFECTS

If 1 lb. of cold water be heated in a closed vessel till the water becomes warm, although the temperature of the water has changed, its *weight* remains the same ; and if the heat be continued until all the water is converted into steam, provided none of the steam can escape, the total *weight* of the steam is still exactly the same as that of the water from which it was produced.

It is evident, therefore, that the *heat* which produced these changes is without weight. Heat cannot, therefore, be a material substance. It was formerly thought to be some kind of subtle fluid, which flowed from hot bodies into colder ones ;

but this theory is now no longer accepted, because it was found that heat could be developed to an unlimited extent from cold bodies merely by rubbing them together.

A piece of cold iron can be made red hot by hammering it. A carpenter's saw, an engineer's chisel, or turning tool, soon get hot when a rubbing action, or friction, is set up between the tool and the work, although they are all quite cold to begin with.

Sir Humphry Davy melted two blocks of ice by rubbing them one upon another, from which he concluded that 'the immediate cause of the phenomenon of heat is *motion*'; and this is now the generally accepted view of the nature of heat.

Still we know that things may be hot without being visibly in motion; hence, if heat is motion, the motion must exist in parts of the body too minute to be seen.

All bodies are composed of minute particles called molecules, held together by mutual attraction or cohesion, and these molecules are in a state of continual agitation or vibration. The hotter the body the more vigorous the vibrations of its constituent particles. In solid bodies the vibrations are limited in extent. If this limit is exceeded, owing to addition of heat, cohesion is sufficiently overcome to enable the particles to move about freely and without restriction, and the solid has now become a liquid. On still continuing the heat, further separation of the molecules takes place, cohesion is completely overcome, and they fly off in all directions. The liquid has now become a gas.

The pressure exerted by the gas on the interior surface of the vessel in which it is confined is due to the collision of the molecules with the sides of the vessel. The greater the intensity of the heat the more violent the impact, and therefore the greater the pressure exerted. This is the condition of things in the interior of a steam boiler.

If a part of the enclosing vessel were movable, it would evidently be pushed backward and outward. This is what happens to the piston of the steam engine.

From what has been just stated, we see that heat is a form of energy, and that heat and mechanical work are mutually

convertible the one into the other. We shall presently show that an exact and invariable relation exists between heat and work.

TEMPERATURE

The *temperature* of a body indicates how hot or how cold the body is, or the *intensity* of the heat of the body.



FIG 1.

The *temperature* of a body should be distinguished from the *quantity* of heat in the body. For example, if a cup of water be dipped out of a pailful of water, the *temperature* of the water is the same throughout, but the *quantity* of heat varies as the weight of water in each vessel.

Thermometers are used to indicate temperature, and they do so by the rise or fall of a little column of mercury enclosed in a tube of very fine bore, and having a small bulb at the bottom containing a store of mercury.

If the thermometer be warmed, the mercury expands or tends to occupy a larger volume, and the column therefore rises in the stem of the tube; or, if the thermometer be cooled, the mercury will contract or diminish in volume, and the column will shorten or fall. A graduated numbered scale is affixed, and the smallest change of temperature, shown by the movement of the surface of the column, is thus easily detected.

To graduate the scale the thermometer is placed in melting ice, and the point to which the mercury in the stem has fallen is marked, and called the *freezing point*. It is then placed in boiling water at the pressure of the atmosphere, and the level of the column of mercury is again marked, and called the *boiling point*.

The distance between these two marks is divided on the Fahrenheit thermometer into 180 equal parts or degrees; on the Centigrade thermometer the distance between the two marks is divided into 100 equal parts or degrees; and on the Réaumur scale the same distance is divided into 80 equal parts or degrees. English engineers mostly use the

Fahrenheit scale. The following sketch will show that the difference between the various thermometers is not in the height of the mercury, but in the scales of degrees by which the height is expressed.

It will be seen from the fig. that the scales are marked as follows :

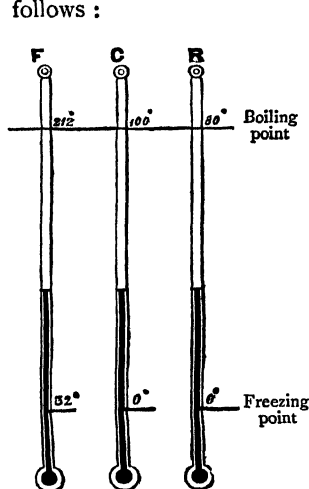


FIG 2.

	Freezing point.	Boiling point.
Fahrenheit . .	32	212
Centigrade . .	0	100
Réaumur . .	0	80

It will also be clear that $212° - 32° = 180°$ F. occupy the same space as 100° C., or 80° R.

$$\text{Now } 100° \text{ C.} = 180° \text{ F.}$$

$$\therefore \quad 1° \text{ C.} = \frac{180°}{100} \text{ F.}$$

$$= \frac{9°}{5} \text{ F.}$$

$$\text{also } 180° \text{ F.} = 100° \text{ C.}$$

$$1° \text{ F.} = \frac{100°}{180} \text{ C.}$$

$$= \frac{5°}{9} \text{ C.}$$

From which we obtain the Rules.—*To convert degrees Fahrenheit into degrees Centigrade :*

Subtract 32, multiply the remainder by 5, and divide by 9.

Thus, convert 158° F. to degrees C.

$$\text{Then } (158 - 32)\,\frac{5}{9} = 70° \text{ C.}$$

Or, *to convert degrees Centigrade into degrees Fahrenheit :*

Multiply by 9, divide by 5, and add 32.

Thus, convert 70° C. into degrees F.

$$\text{Then } (70 \times \frac{9}{5}) + 32 = 158° \text{ F.}$$

The relation between degrees Fahrenheit and Centigrade may also be expressed thus :

$$C. = (F. - 32)\ \frac{5}{9};$$

or, conversely,

$$F. = \frac{9}{5}\ C. + 32.$$

Similarly, the student will see from fig. 2 that 180° F. occupy the same space as 80° R.; hence 1° F. $= \frac{180}{80}$ R. $= \frac{9}{4}$ R.

Also, since 100 C. = 80 R. ∴ 1 C. $= \frac{4}{5}$ R.

Temperatures are reckoned from zero both above and below that point. Temperatures below zero are marked with the negative sign; thus $-25°$ reads minus 25 degrees, and indicates 25 degrees below zero.

Specific Heat

The ratio of the amount of heat required to raise the temperature of a substance one degree to the amount of heat required to raise an equal weight of water one degree is called the *specific heat* of the substance.

The specific heat of bodies varies very considerably, as will be seen from the following table:

I. *Table of Specific Heats*

Water	=1·000
Cast Iron	=0·130
Steel	=0·118
Wrought Iron	=0·113
Copper	=0·100
Bismuth	=0·031
Lead	=0·031
Mercury	=0·033
Coal	=0·241

Water has the highest specific heat of any substance (except hydrogen), and the metals have the lowest. In other words, it takes more heat to raise the temperature of a given weight of

water one degree than to raise the same weight of any other substance one degree. The specific heat of water is 1. The specific heat of wrought iron by the table is 0·113, or about $\frac{1}{9}$; that is to say, the quantity of heat which would raise 1 lb. of wrought iron through 1° F. would only raise the temperature of 1 lb. of water through about $\frac{1}{9}$° F.

The following experiment may be easily performed:

A mass of iron weighing 1 lb. is immersed in boiling water; the iron is raised to the temperature of the water, namely, 212° F., and is then immersed in 2 lbs. of water at 50° F. The temperature of the water can now be taken by a thermometer.

To find the resulting temperature, *t*, of the mixture by calculation:

Note.—The total heat per lb. in any given substance is found by multiplying its temperature by its specific heat (see table, p. 5). Then

$$\text{Heat lost by iron} = \text{Heat gained by water};$$
$$(212-t) \times \text{sp. ht.} \times \text{weight} = (t-50) \times \text{sp. ht.} \times \text{weight};$$
$$(212-t) \times \cdot 113 \times 1 = (t-50) \times 1 \times 2;$$
$$23 \cdot 956 - \cdot 113\, t = 2\, t - 100;$$
$$t = 58 \cdot 66^\circ \text{ F.};$$

that is, the temperature of the mixture is 58·66° F.

Absolute Temperature

The zero of temperature on the Centigrade and Fahrenheit scales has been chosen arbitrarily, on one the zero being the freezing point of water, and on the other a point 32° F. below it.

For scientific purposes it is necessary to have a uniform zero, and such a point, called the zero of absolute temperature, has been chosen (from considerations explained in the Advanced Series), the position of which is 461° F. below the zero Fahrenheit, or 273° C. below the zero Centigrade.

Hence, to express degrees Fahrenheit in degrees of absolute temperature, add 461. Thus the boiling point of water at atmospheric pressure = 212° F. = 212 + 461 = 673° absolute temperature.

CHAPTER II

UNIT OF HEAT AND UNIT OF WORK

BEFORE quantities of heat can be measured, we must have a unit of heat, just as we require a unit of length, namely, the inch or foot, in order to measure distance; or the pound or ton, in order to measure weight.

The unit of heat is the amount of heat necessary to raise the temperature of 1 *lb. of water* 1° *F.*, when the water is at its greatest density, namely, from 39° to 40° F.

But the all-important point with the engineer is the conversion of heat into *work*. We will therefore now consider what is understood by work, how it is measured, and what the relation is which exists between the two.

By the term *work* in mechanics is understood 'the overcoming of a resistance through a space,' and the amount of work done is measured by the resistance overcome, multiplied by the distance through which it is overcome, the resistance being measured in pounds, and the distance in feet.

Thus, if a body weighing 7 lbs. be lifted through a height of 3 feet, then the resistance, namely, 7 lbs., multiplied by the distance through which it is overcome, namely, 3 feet, is equal to $7 \times 3 = 21$ foot-pounds of work. Hence, work is measured neither by the pound nor by the foot, but by the product of the two. Thus the *unit of work* is the work done in raising one pound through a vertical height of one foot, and is called the *foot-pound.*

Or, since action and reaction are equal and opposite, we may consider the *force* which overcomes the resistance. The work done by a force is measured by the intensity of the force,

multiplied by the distance through which it acts, measured in the direction of the force. Thus, as before, in the above example, a force of 7 lbs. overcame the resistance due to the weight, and acted through a space of 3 feet, doing thereby $7 \times 3 = 21$ foot-pounds of work.

Since the unit of work is a product of two numbers, it may be represented by an area, and this is important, for we intend by-and-by to estimate the work done by an engine from the area of an indicator figure. Thus, if $\frac{1}{8}$ inch be taken to represent pounds on one line, and $\frac{1}{4}$ inch to represent feet on a line at right angles to it, then the unit of work is given by the small cross-lined rectangle, and the 21 foot-pounds in the above example by the whole rectangle (fig. 3).

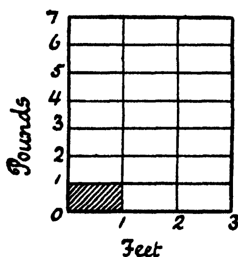


Fig. 3.

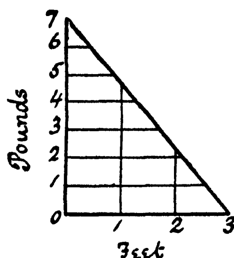


Fig. 4.

Again, suppose the weight lifted in the previous case to be a vessel containing 7 lbs. of water, and that the water should escape by a tap at a uniform rate, all the while it is being lifted, until, when a height of 3 feet is reached, there is no water left. This result also may be well shown by a diagram (fig. 4), where the weight, varying from 7 lbs. to nothing, is given by a diagonal falling from 7 to the zero line of weight.

The total work done is again given by the area of the figure, and is evidently equal to the distance 3 multiplied by the *mean* weight $= 3 \times \frac{7+0}{2} = 3 \times 3\frac{1}{2} = 10\frac{1}{2}$, or one-half that in the previous case.

It should be noticed that the unit of work has no reference

to the *time* taken, for the same amount of work is done in lifting the weight, whether it be done in one second or one hour.

The *power* of an agent is measured by the *rate* at which it can do work, and depends upon the amount of work done in the unit of *time*.

The unit of power adopted by engineers is the *horse-power*.

A horse-power represents the performance of 33,000 foot-pounds of work per minute. The addition of the words *per minute* should be particularly noticed.

$$\frac{\text{Work done}}{\text{Time in minutes}} = \text{units of work done per minute;}$$

$$\text{and } \frac{\text{Work done}}{33{,}000 \times \text{time in mins.}} = \text{horse-power exerted.}$$

Energy is defined as 'the power of doing work.' When heat is applied to water, it confers upon the steam which is produced the power of doing work, such as driving the piston from one end of the cylinder to the other against a resistance; and if we take the case of the locomotive, for example, the heat energy in the boiler furnace is capable of being transformed into the energy of motion of the moving train.

If the brakes are put on the moving train, then the energy of motion of the train is retransformed into heat, sparks fly from the wheels and rails, and the train is brought to a standstill.

It is a fundamental principle in nature that, just as matter can neither be created nor destroyed, though it may be made to assume different forms, visible or invisible, so energy, whether heat energy or any other, cannot be destroyed. It may take a variety of different forms, but the sum total of the energy remains the same. This principle is called the principle of the *conservation of energy*.

Hence the heat which is carried to the engine in the steam is either transformed into useful work, or it passes away to waste in various ways, and the sum of the heat usefully employed plus the heat which is wasted always equals exactly the heat which was applied.

MECHANICAL EQUIVALENT OF HEAT

We may now consider the important question of the relation between the unit of heat and the unit of work.

The following diagram (fig. 5) will give an approximate idea of the apparatus used by Joule to determine this relation.

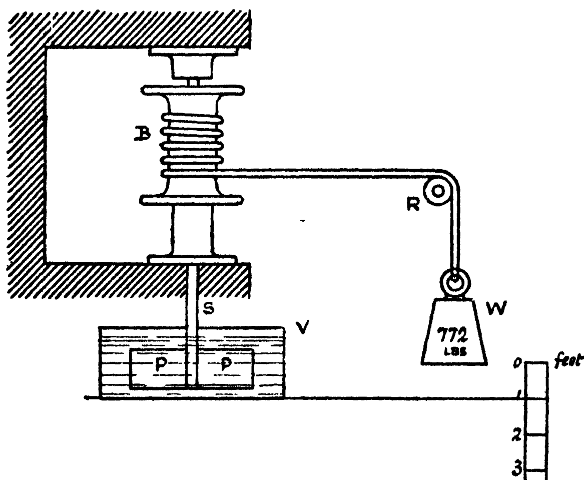


FIG. 5.

The weight W is attached to a string which is wound round the barrel B. A spindle S passes through the barrel, having thin pieces of sheet metal P P forming paddles or vanes attached to and radiating from it. The paddles are immersed in a vessel of water. When the weight W falls, the paddles rotate in the water, the water itself being prevented from rotating by fixed pieces not shown.

When the weight descends one foot, 772 foot-pounds of work have been done, for the weight could have lifted an equal weight at the other end of the string. This work, which cannot be lost, now appears as heat in the water, the agitation of the paddles having increased the temperature of the water by an amount which can be measured by the thermometer.

By this method (here only roughly indicated) Dr. Joule determined with the utmost care that 1 lb. of water was increased in temperature one degree by the work done upon it during the descent of 772 lbs. through 1 ft. Hence

1 unit of heat=772 units of work.

To convert units of heat into units of work: Multiply the units of heat by 772.

And *vice versâ*, to convert units of work into units of heat: Divide the units of work by 772.

If the unit of heat be measured by the Centigrade scale, namely, the heat necessary to raise the temperature of 1 lb. of water 1° C., then substitute the number 1390 for 772; for

$$772 \times \frac{9}{5} = 1390 \text{ nearly.}$$

The above experiment establishes the relation between heat and work by converting work into heat.

The business of the mechanical engineer consists of constructing machines by means of which the converse process, namely, the conversion of heat into work, may be carried out; and the result of a large number of engine trials goes to prove conclusively the truth of the relation between heat and work as established by Joule, namely, that one unit of heat=772 units of work. A horse-power expressed in thermal or heat units $=\frac{33000}{772}=42 \cdot 75$.

CHAPTER III

TRANSFER OF HEAT

WHEN bodies of unequal temperature are placed near each other, the hot body tends to part with its heat to the colder body until the temperature in each is equal; and when there is no tendency to a transfer of heat between them they are said to be of equal temperature.

The transfer of heat from one to the other may take place in any of the following ways: namely, by radiation, conduction, or convection.

RADIATION

Heat is given off from hot bodies in rays which radiate in all directions in straight lines. The heat from the burning coal in a furnace is transferred to the crown and sides of the furnace by radiation; it passes through the furnace plates by conduction, and the water is heated by convection.

CONDUCTION

The process by which heat passes from hotter to colder parts of the same body, or from a hot body to a colder body in contact with it, is called *conduction*. A bar of iron having one end placed in the fire soon becomes hot at the other extremity, the heat being conducted from particle to particle throughout its entire length.

A piece of burning wood can be held with the hand close to the burning part. Evidently, therefore, some bodies conduct heat much more readily than others.

If a piece of clean paper be pasted on the bottom of a copper kettle containing water, and the kettle be placed on a bright fire or over a strong gas flame, the water will soon be warmed, but the paper will not be charred in the least; the reason of this being that the heat is so rapidly conducted by the copper to the water. Bodies which conduct heat readily are called good conductors; those which conduct heat slowly are called bad conductors.

Bad conductors are used by engineers to prevent loss of heat by radiation; hence boilers, steam pipes, and cylinders are covered with some non-conducting material, such as hair felt, or asbestos. Bodies of a finely fibrous texture are the worst conductors of heat.

The following table gives the relative conducting power of metals:

Silver,	100	Steel,	11·6
Copper,	74	Lead,	8·5
Brass,	23	Bismuth,	1·8
Iron,	11·9		

Liquids and gases are bad conductors, and it is impossible to heat them by conduction; but they may be very readily and quickly heated by convection.

Convection

Convected or carried heat is that which is transmitted from one place to another by currents. The following experiment will clearly show that water is a bad conductor, and the necessity therefore of heating it by some other method than conduction. Take a test tube nearly full of cold water, and hold the tube with the upper surface of the water against a flame, as shown in fig. 6. The water will soon boil at its upper surface, while the temperature of the water in the bottom of the tube is not appreciably changed; for, if a piece

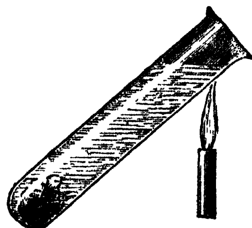


Fig. 6.

of ice be placed in the bottom of the tube, it will remain un-melted. If, however, the heat be applied at the bottom of the vessel, fig. 7, the heated lower layers, becoming less dense, rise towards the surface, while the cold upper and denser layers

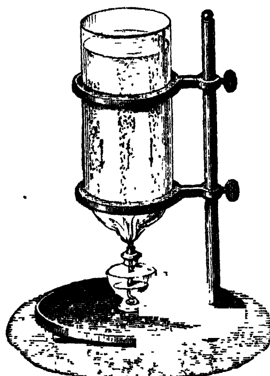


FIG. 7.

fall, and thus circulating currents are set up which can be very plainly seen by dropping a little bran into the water, and which soon result in the water being heated throughout.

Steam boilers should be so constructed as to secure the ready circulation of the water.

CHAPTER IV

COMBUSTION OF FUEL

THE principal fuel used by engineers is coal, and the chief constituents of coal are carbon and hydrogen.

Atmospheric air consists of two invisible gases, oxygen and nitrogen, in the proportion of 23 parts of oxygen and 77 parts of nitrogen in every 100 parts of air by weight. These gases are not united in any way, they are merely mixed together. The oxygen is the active element in air, and it is ready to unite with anything for which it has affinity, providing the surrounding temperature is raised sufficiently high to enable it to do so. All fuels contain elements which readily unite with oxygen. The nitrogen of the air takes no part whatever in the process of combustion, and merely serves to dilute the oxygen. The process of combustion may be easily understood by considering the case of the common gas flame in the house.

When we wish to 'light the gas'—that is, to set in operation the process of combustion, or chemical union between the oxygen of the air and the carbon and hydrogen of the gas—we have first of all to apply heat with a match; otherwise, if the tap is turned on, the gas will escape, but it will not burn. Once started, however, the burning proceeds vigorously and uniformly, and results in the evolution of heat. Before the escaping gas was lighted we could detect the strongly characteristic odour of unburnt coal gas, but no such odour can be detected from the burning gas flame. The reason of this is that the carbon and hydrogen of the coal gas have united with the oxygen of the air to form two odourless and invisible compounds, namely, carbonic acid gas (CO_2) and steam (H_2O).

In such a gas flame, however, the combustion is not per-

fect, owing to the incomplete mixture of the coal gas with the oxygen of the air ; hence the ceiling of the room is eventually blackened by the deposit of unburnt carbon.

The combustion of coal differs, however, from the case just considered ; for, when coal is thrown on a furnace, there are three distinct stages in its combustion : first, the gases contained in the coal are distilled off as in the ordinary process of gas making ; secondly, these gases are either consumed or pass up the chimney unconsumed ; thirdly, the remaining solid residue of the coal is burnt. Considering the gases distilled from the coal, which consist principally of marsh gas (CH_4) and olefiant gas (C_2H_4) : in order that they may be completely burnt, (1) they must be thoroughly mixed with a sufficient supply of oxygen ; hence the necessity of admitting air above the coal, not, however, in excess, otherwise our object would be defeated by the cooling of the furnace. (2) The temperature of the mixed gases must be sufficiently high to allow of chemical combination taking place.

When the distilled gases from the coal are not mixed with a sufficient supply of oxygen, or the temperature is not sufficiently high, then large quantities of carbon are disengaged from the gas, and pass up the chimney in the form of smoke, part of which is deposited in the flues as soot. But if the disengaged carbon is supplied with sufficient oxygen, and the temperature is sufficiently high for ignition or combination to take place, it burns with a bright flame.

Considering the solid fuel which remains as coke or carbon, it should be explained that carbon is capable of forming two different compounds with oxygen, namely, carbonic oxide (chemical symbol, CO) and carbonic acid gas (chemical symbol, CO_2), depending on the abundance of the supply of oxygen to the carbon during the process of combustion.

When the supply of oxygen is sufficient, and is intimately mixed with the fuel, the carbon is completely burnt to carbonic acid (CO_2); but when there is an insufficient supply of oxygen, or the oxygen is not intimately mixed with the fuel, then carbonic oxide (CO) is formed. The effect of this on the production of heat may be seen by the following table :

II. *Table of Heat of Combustion*

Combustible.	Total units of heat of combustion per lb.	lbs. of water evaporated from and at 212°.
Hydrogen	62,032	64·2
Carbon burned to carbonic oxide	4,400	4·55
Carbon burned to carbonic acid	14,500	15·0
Anthracite	14,700	15·2
Newport coal . . .	14,000	14·5
Durham coke . . .	13,640	14·1
Wigan cannel coal . . .	14,000	14·5
Petroleum	20,360	21·0
Coal gas	20,800	21·5
Oak wood (dried) . . .	7,700	8

When the air for combustion in a boiler furnace passes between the fire-bars under the fuel, combination takes place between the oxygen and the under layers of glowing carbon, forming carbonic acid (CO_2). This gas, in passing on through the upper layers of carbon, here loses part of its oxygen, and the carbonic acid gas (CO_2) is now reduced to carbonic oxide (CO); the remainder of its oxygen having united with more carbon to form carbonic oxide also.

If now sufficient air is supplied at the surface of the fuel, this carbonic oxide will burn with a blue flame, with further evolution of heat; but if it is not so supplied, it will pass up the chimney unconsumed, and the difference between the heat of complete and incomplete combustion of carbon, namely, 10,100 units of heat per lb. of carbon, will be lost.

Example.—At a recent engine trial the coal burnt in the boiler furnace had a calorific value of 14,200 thermal units per lb.; the weight of coal used per hour was 40·7 lbs.: find the total number of thermal units per hour to be accounted for. *Ans.* 577,940 thermal units.

One pound of good coal yields (when the combustion is

perfect) 14,000 units of heat; or, $14,000 \times 772 = 10,808,000$ units of work, or approximately 10,000,000 units of work.

A *horse-power* is equal to 33,000 units of work done per minute; or, $33,000 \times 60 = 1,980,000$ units of work per hour; or, roughly, 2,000,000 units of work per hour.

Hence the heat from 1 lb. of coal, if all utilised, would be capable of exerting an energy of $\frac{10,000,000}{2,000,000} = 5$ horse-power per hour (approximately), or, in other words, $\frac{1}{5}$ lb. of coal might be expected to yield one horse-power per hour. But our best stationary engines consume 2 lbs. of coal per horse-power per hour, or ten times as much as would be necessary if the whole of the heat of the coal could be converted into work. This, however, can be shown to be impossible, and the ideal performance beyond which the steam engine is never likely to go is a consumption of about 1 lb. of coal per indicated horse-power per hour. Triple expansion marine engines now consume about $1\frac{1}{2}$ lb. of coal per indicated horse-power per hour.

Evaporative Power of Fuel

It will be shown in a future chapter that the amount of heat required to convert 1 lb. of water at 212° F. into steam at 212° is equal to 966 units. Hence, if we know the number of units of heat obtained by the complete combustion of 1 lb. of fuel, we can find the number of pounds of water at 212° which this heat will convert into steam at the same temperature, or, in other words, the *evaporative power* of the fuel.

Example 1.—Find the evaporative power of 1 lb. of pure carbon.

$$\frac{\text{Total heat evolved per lb.}}{\text{Heat required per lb. of water}} = \frac{14,500}{966} = 15 \text{ lbs. of water.}$$

Example 2.—Check the accuracy of the results given in last column of Table II.

The values given in the table of the evaporative power of fuels assume that the combustion is perfect, that the water is at 212° F., and that the whole of the heat goes to evaporate

water. This, however, is not the case in actual practice, for the heat in the boiler furnace is expended in

(1) heating the water in the boiler and converting it into steam;

(2) heating the furnace gases, which escape by the chimney, carrying their heat with them to waste;

(3) heating surrounding objects by radiation;

(4) evaporating the moisture in the coal.

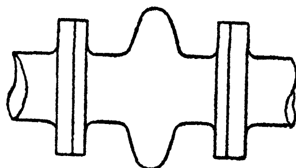
The evaporative power of the best types of steam boilers at the present time is about 9 to 10 lbs. of water per lb. of coal.

At a recent engine trial a compound portable boiler evaporated 18·6 lbs. of water for an expenditure of 1·86 lbs. of coal per indicated horse-power per hour. This is equivalent to 10 lbs. of water evaporated per lb. of fuel consumed.

CHAPTER V

APPLICATION OF HEAT TO SOLIDS

ALL bodies expand by the action of heat. Numerous examples of the application of this law of expansion of metals will occur to students of engineering. Thus the bars of boiler furnaces are left free at the ends to enable them to expand. Boiler plates are riveted with red-hot rivets which cool and contract and draw the plates together at the joint with great force. In laying railways, a small space is left between successive lengths of rail; and the bolt-holes by which they are secured to the fish-plates are elongated. Tires of wheels are fitted on when red hot, and as they cool they contract and grip the wheel with great firmness. Cranks are 'shrunk on' crank shafts in a similar way. The walls of buildings which bulge out in the centre have been drawn back into position by passing iron bars through the walls from side to side of the building. They are screwed at the end with nuts and have large plate washers. The bars are heated inside the building and the nuts are tightened up. On cooling, the bars contract and draw the bulged walls together. Steam pipes, which are rigidly secured between two cylinders, should be fitted with an expansion joint or connection. A flanged copper connecting pipe, which admits of being extended or compressed, as in fig. 8, is sometimes used.


FIG. 8.

Engine cylinders, which are heated to the temperature of the steam, instead of being rigidly bolted down on a horizontal

bed-plate, are frequently, especially for high-pressure steam, secured by the front face, the rest of the cylinder overhanging the bed of the engine. A small space is allowed between the crank bosses and main bearings of engines having cast-iron bed-plates to allow of expansion of the crank shaft in case of hot bearings, &c. If glass is heated or cooled suddenly, it is very liable to crack, because glass conducts heat slowly, and the two sides of the glass are unequally heated, and therefore unequally expanded: hence the fracture. The same thing is liable to occur in steam cylinders, which should always be carefully warmed by opening the stop valve a little while the steam is being generated, and blowing gently through with steam for some time before starting the engines, and thus bringing the cylinders and jackets gradually up to the working temperature.

Steam boilers also require great care for similar reasons. They should not be hurriedly heated or cooled, and all sudden changes of temperature should be avoided; otherwise, unequal contraction will take place, resulting in leakages.

Less harm is done to a boiler by steaming steadily for a length of time than by repeatedly getting up steam and drawing the fires, which brings about repeated expansions and contractions of the boiler.

The force exerted by heat in expanding a bar of metal is the same as would be required to stretch it to the same extent by mechanical means.

Application of Heat to Gases

Gases, such as air, expand on the action of heat much more freely than liquids or solids. The law which expresses the behaviour of gases under the influence of heat is known as the Law of Charles, and it may be stated thus:—The volume of a gas under constant pressure, or the pressure of a gas at constant volume, varies as the absolute temperature. The meaning of this law will be clear on considering the following applications.

Example 1.—A quantity of air in a cylinder under a movable piston

occupies 10 cub. ft. at 60° F.; what volume will it occupy if heated to 250° F. under the same constant pressure?

Here, the volume occupied by the air will evidently be greater, and in proportion to the absolute temperature, thus:

$$60° \text{ F.} = 60 + 461 = 521 \text{ absolute temperature}$$
$$250° \text{ F.} = 250 + 461 = 711 \quad ,, \quad ,,$$

$$\text{Then, vol. at } 250° \text{ F.} = \text{vol. at } 60° \times \frac{711}{521}$$
$$= 10 \times \frac{711}{521}$$
$$= 13{\cdot}65 \text{ cub. ft.}$$

Example 2.—A volume of air at 212° F. is confined in a rigid cylindrical vessel, and exerts a pressure of 15 lbs. per square inch; find the pressure exerted by the air when the temperature is increased to 300° F., the volume, of course, remaining the same.

Here, by the above law, the pressure exerted by the air will be greater, and in proportion to the absolute temperature; then,

$$212° \text{ F.} = 212 + 461 = 673 \text{ absolute}$$
$$300° \text{ F.} = 300 + 461 = 761 \quad ,,$$

$$\text{Then pressure at } 300° \text{ F.} = \text{pressure at } 212 \times \frac{761}{673}$$
$$= 15 \times \frac{761}{673}$$
$$= 16{\cdot}96 \text{ lbs. per sq. in.}$$

If the temperatures are given in degrees Centigrade instead of Fahrenheit, then to find the absolute temperature add 273, (see p. 6) thus:

Example 3.—A certain quantity of gas occupies 20 cubic feet at 15° C., what volume will it occupy if its temperature is raised to 100° C., the pressure on the gas remaining constant?

$$15° \text{ C.} = 15 + 273 = 288 \text{ absolute}$$
$$100° \text{ C.} = 100 + 273 = 373 \quad ,,$$
$$\text{then } 20 \times \frac{373}{288} = 25{\cdot}9 \text{ cub. ft.}$$

PRESSURE OF THE AIR—ABSOLUTE PRESSURE

On the surface of the earth we live, as it were, at the bottom of an aërial sea, which we call the atmosphere, and its weight causes a pressure in every direction of 14·7 lbs., or, roughly, 15 lbs. per square inch.

Pressures are usually reckoned from the pressure of the

atmosphere. Thus the boiler pressure gauge, when its finger points to 10 lbs., indicates a pressure of 10 lbs. *above the atmospheric pressure.* To express this in *absolute pressure* add the pressure of the atmosphere to the gauge pressure.

Thus, 10 lbs. pressure by boiler gauge $= 10 + 15 = 25$ lbs pressure absolute.

APPLICATION OF HEAT TO WATER

Water is a compound substance, consisting of hydrogen and oxygen chemically combined in the proportion of two volumes of hydrogen to one volume of oxygen, written in chemical symbols H_2O.

When water is subjected to the action of heat it is converted into *steam*, whch is water in the gaseous state.

Though a change thus takes place in the physical condition of the substance, the chemical composition of the steam is in no way different from that of the water from which it is generated.

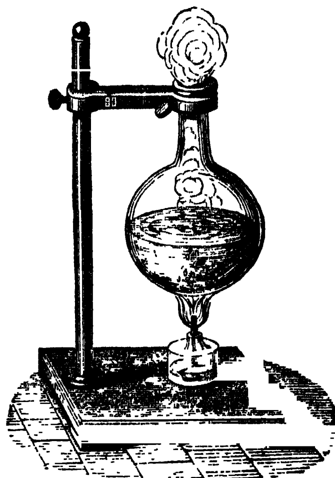


FIG. 9.

BOILING

If heat be applied to the bottom of a vessel, as in fig. 9, the air contained in the water will first appear as little bubbles which rise to the surface. Then the water immediately in contact with the source of heat will be converted into steam. The steam will form as bubbles on the bottom, and these will rise through the liquid ; but at the commencement of the operation they will at once be condensed by the cold upper layers of water. The condensation of the bubbles of steam is the cause of the

'singing' of the water before boiling. Finally, the water becomes heated throughout until it reaches a temperature of 212° F. under *the pressure of the atmosphere*, when the bubbles rise to the surface and boiling begins.

It should be particularly noted that the temperature at which boiling takes place depends upon the pressure on the liquid, and that for every different pressure there is a fixed temperature at which boiling takes place, so that water has an indefinite number of boiling points.

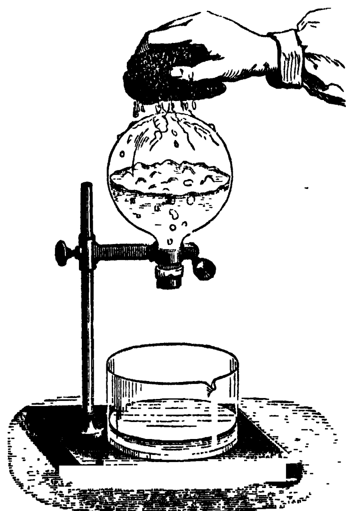


FIG. 10.

An experiment illustrating boiling at a low temperature will be understood by reference to fig. 10. Water is boiled in a glass flask as in fig. 9. When the water has been boiling a little time, and all the air is expelled, the heat is removed, and the flask is closed by a cork, turned upside down, and placed on the stand as shown. Meantime the water has, of course, ceased to boil. If now cold water be poured gently on the flask, the steam which occupies the space above the water will be condensed, *the pressure on the water will therefore be reduced*, and the water will again boil vigorously, although the temperature of the water has by this time fallen considerably below 212° F.

Similarly, owing to the reduced pressure of the atmosphere on the tops of high mountains, boiling water is not sufficiently hot to cook food. On the other hand, the temperature of boiling water at the bottom of deep mines is higher than at the surface.

The boiling temperatures for water under varying pressures are given in Table III., p. 39. The following are a few important pressures and temperatures :

Under a pressure of		5 lbs.	the boiling temp. is		162° F.
"	"	10	"	"	193° "
"	"	1 atmosphere		"	212° "
"	"	2 atmospheres		"	249° "
"	"	3	"	"	273° "
"	"	4	"	"	291° "
"	"	5	"	"	306° "
"	"	10	"	"	357° "

The presence of solid bodies such as salt dissolved in the water raises the temperature of the boiling point. Thus the boiling point of sea water under atmospheric pressure is 213·2° F.

CONDENSATION OF STEAM—VACUUM

Steam is water in the gaseous condition, and when the steam is cooled, it again returns to the liquid state and becomes water.

Thus, let a flask A contain a known weight of pure water. Fit a cork and glass tube to it as shown, and connect with a spiral tube surrounded by flowing cold water; let the lower end of the tube pass into a vessel B. Boil the water in A. It will pass off as steam by the tube C to the spiral; and if the spiral be surrounded by a stream of cold water, the steam will be condensed to water, which will drop from the end of the tube.

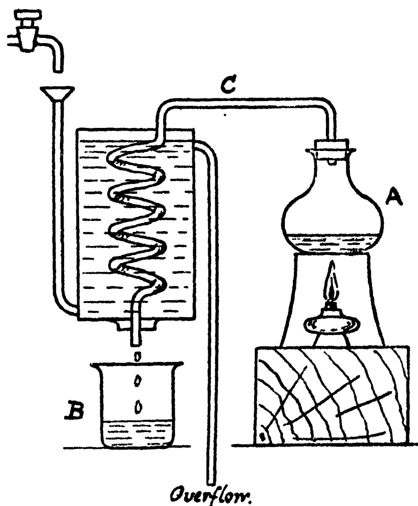


FIG. 11

At the end of the operation the loss of weight by A is equal

to the gain by B. This illustrates the process of distillation, and by this method pure water may be obtained from water containing impurities.

Advantage was taken by the early engineers of the property possessed by steam of being easily condensed. They valued steam not so much for its own sake, but because by condensation they were able to call to their aid the pressure of the atmosphere in the performance of work.

A *vacuum* is literally an *empty space*—that is, a space absolutely free from air or vapour of any kind capable of exerting pressure.

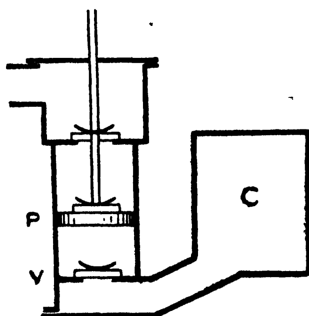


FIG. 12.

Vapour arises from water at *all* temperatures, and exerts an appreciable pressure. And the lowness to which the pressure can be reduced in condensers depends on the temperature of the condensed steam, and this temperature in practice cannot economically be reduced below about 102° F., at which temperature the vapour of water exerts a pressure of 1 lb. per square inch.

But, further, the condensed steam, vapour, and air in engine condensers are removed by a pump called an *air pump*, as in fig. 12.

Now, when the plunger or pump bucket P is lifted, the valve V will lift by virtue of the difference in pressure on the two sides of the valve. Assuming that we could obtain a perfect vacuum in the pump chamber, yet the pressure per square inch in the condenser C can never fall below that necessary to lift the valve V.

Experiment.—Take a thin tin cylinder closed at both ends, having a tap, *t*, at one end. Pour a little water into the cylinder by the tap. The vessel now contains air and water. Boil the water till the steam escapes from *t* and has driven most of the air out. Now the vessel contains steam and very little air.

Close the tap and pour cold water on the vessel. The steam is immediately condensed to water; and since water occupies only about $\frac{1}{1650}$ of the space of the steam at atmospheric pressure, a partially empty space has been formed inside the vessel, and the external pressure of the atmosphere will collapse or crush the vessel.

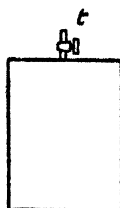


Fig. 13.

If the cylinder had been made strong enough to resist the excess of external pressure over internal pressure, and a tube had been led from the cylinder into water some depth below it, then the water would be forced up the tube into the cylinder by the pressure of the atmosphere, till the pressure on the inside of the cylinder is the same as the atmospheric pressure outside. Here, then, useful work would be done in lifting water from a low level to a higher level, and this was the principle of the early pumping engines as made by Savery. Again, if the top of the cylinder had been movable, then it would act like a piston, and be forced towards the bottom. This was the principle of Newcomen's engine, which was called the 'atmospheric' engine, because the work was really done by the atmosphere on the piston after a vacuum had been formed in the cylinder by the condensed steam.

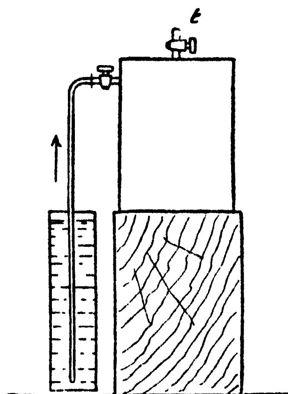


Fig. 14.

CHAPTER VI

ACTION OF HEAT IN THE FORMATION OF STEAM

THE action of heat in the formation of steam from water may be illustrated by the following diagrams.

(1) Let the cylinder (stage 1, fig. 15) contain 1 lb. of water at 32° F., and let the pressure of the atmosphere be

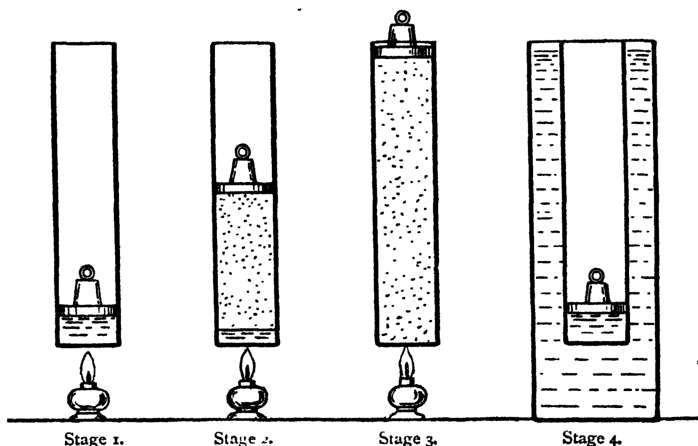


FIG. 15.

represented by a weighted piston. Then, if heat be applied to the water, the temperature will rise higher and higher, though the piston will remain stationary, except for the small expansion of the water, until the temperature of the water reaches 212°.

(2) On continuing the heat the water shows no further increase of temperature by the thermometer, but steam begins to form and the piston commences to ascend in the cylinder (stage 2);

rising higher and higher as more and more steam is formed, until the whole of the water is converted into steam. In stage 1 the steam did not begin to form until the temperature reached 212°. Evidently, therefore, this is the lowest temperature at which steam can exist under atmospheric pressure.

Again, in stage 3, as soon as the last drop of water disappears, we have 1 lb. of steam occupying the least possible volume at the given pressure; the steam in this condition is termed *saturated* steam.

(3) If the heat is continued the steam will become *superheated*—that is, its temperature will rise above that of saturated steam, and the piston will continue to rise.

(4) If the steam be surrounded by a vessel containing an indefinite supply of cold water (stage 4), then the heat will be extracted from the steam by the surrounding water, and the steam will be condensed to water, the same in every particular as to weight and properties as the water with which we started; and if the temperature of the water is now the same as its temperature before starting, then the whole heat taken away when the steam is condensed is equal to the whole heat added during the operation. The series of changes have, therefore, been brought about by the addition or subtraction of heat only.

We have so far been content with a general statement of the action of heat in the formation of steam; we will now consider what *quantities* of heat are required to perform the several stages of the operation.

Work done by Steam during Formation

Referring to fig. 16, let 1 lb. of water at 32° F. be contained at the bottom of a cylinder 1 sq. ft., or 144 sq. ins., in sectional area. Then, first to find the height of the water in the cylinder; since the area of the vessel is 1 sq. ft., and the weight of 1 cubic foot of water is 62·5 lbs.,

62·5 lbs. of water will stand 1 ft. high,

1 lb. " " $\frac{1}{62{\cdot}5}$ ft.

$= {\cdot}016$ ft.

Let the pressure of the atmosphere be represented by a piston resting on the surface of the water loaded with a weight of 14·7 lbs. per sq. in.

The area of the piston being 1 sq. ft., the total weight on the piston is therefore $14·7 \times 144 = 2116·8$ lbs.

(1) On applying heat to the water, it will at first gradually rise in temperature from 32° to 212° before evaporation commences, as explained on page 18.

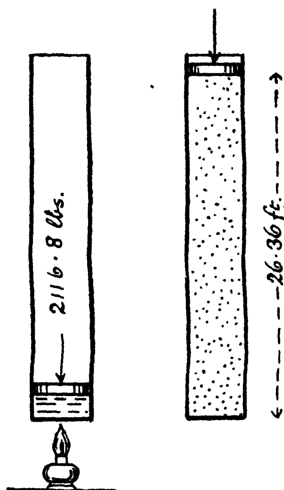


FIG. 16.

Then, $212 - 32 = 180 =$ the number of heat units required to raise water from 32° to boiling temperature at atmospheric pressure, and this represents the heat units expended in stage 1, fig. 15.

(2) Steam now begins to form and the piston to rise; and, on continuing the heat, the water is eventually all converted into steam at 212°, and the piston continues to rise till the steam occupies a volume, under the pressure of the atmosphere of 26·36 cub. ft. as found by experiment (see Regnault's Tables, p. 39).

The heat expended in evaporating the 1 lb. of water at 212° into 1 lb. of steam at 212° is found to be 966 units. Hence the total heat required, first to raise water from 32° to 212°, and then to convert it into steam at the same temperature under atmospheric pressure, $= 180 + 966 = 1,146$ units. Now, in stage 1, fig. 15, it is quite evident how the heat has been expended, namely, in raising the temperature of the water; but in converting the water into steam, though 966 units of heat have been expended, there is no increase in temperature, and it is not at first quite clear what has become of this heat; hence it was called latent or hidden heat. The term 'latent,' however, is not well chosen for the following reasons:

It will be noticed that in this operation two things have happened: firstly, the water has all been converted into steam, which occupies a greatly increased volume (1,644 times, at atmospheric pressure) as compared with the water from which it was generated; and, secondly, the piston has been raised from the surface of the water in stage 1 to that of the steam in stage 3. The heat, usually called latent heat, has been expended, then, in two ways: firstly in overcoming the internal molecular resistances of the water in changing its condition from water to steam; and, secondly, in overcoming the external resistance of the piston to its increasing volume during formation.

The first of these effects of 'latent' heat is called *internal* work, because the changes have been wrought within the body itself; and the second is called *external* work, because the work has been done on bodies external to itself; and these two kinds of work must be carefully distinguished. The first represents energy contained *in* the steam; the second represents energy which has passed out of it, having been expended in doing work on the piston.

We will now consider what share of the heat has been expended on each operation respectively.

The heat expended in doing the external work of raising the piston under a pressure of 2,116·8 lbs. through a height of 26·36 ft. = 2,116·8 × 26·36 = 55,799 foot lbs.; or, 55,799 ÷ 772 = 72·3 units of heat.

Now, the total heat applied to the water, as we have seen, is 1,146 units; and we have so far accounted for 180 + 72·3 = 252·3 units, leaving a difference of 1,146 − 252·3 = 893·7 units, and this difference represents the heat absorbed in doing the internal work of converting the water into steam.

The distribution of the heat may be summarised as follows:

		units.
(1) In raising temp. of water from 32° to 212°	=	180
(2) In overcoming internal resistance	=	893·7
(3) In raising piston against external resistance	=	72·3
Total heat	=	1,146·0

Now, the external work done per lb. of steam during its formation may be represented by an area. For the pressure P

per square foot multiplied by the area of the piston in square feet gives the load on the piston, and this multiplied by the height l through which the piston moves in feet gives the work done; or

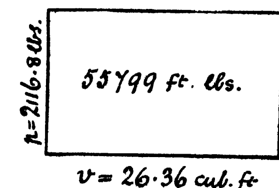


FIG. 17.

$$\text{External work} = P \times a \times l.$$

But $a \times l = v =$ the volume occupied by the 1 lb. of steam; therefore

$$\text{External work} = P \times v.$$

If, then, a rectangle be constructed, as in fig. 17, having one side $= P$, and an adjacent side $= v$, to any convenient scale, the area of the rectangle will equal the work done.

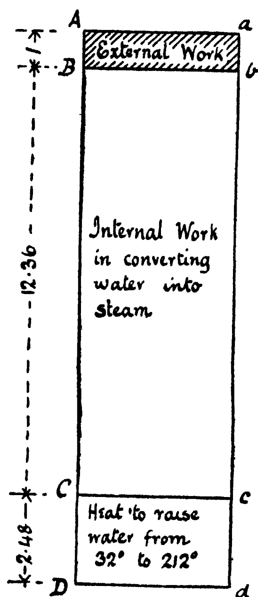


FIG. 18.

Similarly, the proportion which the heat converted into external or useful work bears to the whole heat expended may be shown by the aid of rectangular areas.

From the above summary of results we see that the ratio of the thermal units expended as described is as 180 : 893·7 : 72·3; or, dividing each of the numbers by 72·3, we have 2·48 : 12·36 : 1.

Draw the rectangle A B b a (fig. 18), making A B = pressure and B b = volume to any scale to represent the external work done by the steam. To the base B b add the rectangle B C c b = 12·36 times the rectangle A b. This is done by making B C = 12·36 times A B. Make also C D = 2·48 times A B and complete the rectangle. Then the total heat required to heat 1 lb. of water from 32° to 212°, and to convert it into steam at the same temperature, is given by the rectangle A D d a, and the share of this which goes to perform useful work is represented

by the remarkably small area given by the rectangle A B *b a*. But the ratio which the useful work done bears to the total heat expended is called the *efficiency* of the steam. Hence, in this case, the efficiency $= \frac{\text{area A B } b\, a}{\text{area A D } d\, a} = \frac{1}{15 \cdot 84}$ or about $\frac{1}{16}$.

In other words, in such an engine as this, taking steam at full pressure throughout the whole stroke, only $\frac{1}{16}$ of the heat is usefully employed, while the remainder escapes into the air or condenser in the exhaust steam, except the small part which is wasted by radiation and conduction.

Hence, for every 16 lbs. of coal consumed, the heat from 1 lb. only is converted into work, or $\frac{1}{16} \times 100 = 6 \cdot 25$ per cent. And this is better than would be the case in practice under the same circumstances, because we have neglected the many sources of loss which will be described hereafter.

We may now consider the effect of using steam at a higher pressure than that of the atmosphere. Take, for example, steam at 100 lbs. per square inch absolute.

The external work done by 1 lb. of steam at 100 lbs. pressure per square inch absolute, having given that 1 lb. of steam at 100 lbs. pressure occupies 4·33 cubic feet, is found as follows:

$$P = 100 \times 144 = 14{,}400 \text{ lbs.}$$
$$\text{and } v = 4 \cdot 33 \text{ cub. ft.}$$

Then total external work of steam during formation $= P \times v$

$$= 14{,}400 \times 4 \cdot 33 = 62{,}352 \text{ ft. lbs.}$$

Comparing this with the external work done by 1 lb. of steam at atmospheric pressure, we have

	external work in ft. lbs.
1 lb. steam at 100 lbs. pressure	= 62,352
1 lb. „ 14·7 „	= 55,799

and these numbers do not differ very greatly.

From this we see that, when steam is *not used expansively* —that is, when it is supplied at full pressure throughout the stroke—1 lb. of high-pressure steam is not capable of doing much more useful work than the same *weight* of low-pressure steam.

In comparing the work done by high- and low-pressure steam, it will be noticed we have taken the work done by equal *weights* and no expansion. The same would not be true of equal *volumes*, for evidently if the cylinder were supplied with high-pressure steam, it would do more work on the piston than the same volume of steam at a lower pressure; but then there would be a proportionally greater weight of steam used, and, therefore, a greater quantity of fuel consumed; and the object of the engineer is to get the greatest amount of work from the least consumption of fuel. Thus, if a cylinder is filled at each stroke with steam at 100 lbs. pressure per square inch throughout, then, assuming there is no back pressure, this steam would do twice as much work as steam at 50 lbs.; but the weight of each cylinder full at 100 lbs. pressure is approximately twice that of the cylinder full at 50 lbs. Hence, though we have done twice the work, we have used twice the weight of steam, and, therefore, weight for weight, the work done in both cases is equal.

To find the work done per lb. of steam during formation, without expansion, at any given absolute pressure per square inch p:—Find by Table III. (p. 39) the given pressure, the volume v per lb. in cubic feet; then $p \times 144 \times v =$ work done.

Example.—Find the external work done per 1 lb. of steam at 60 lbs. pressure absolute; then by Table III., vol. per lb. of steam at 60 lbs. pressure = 7·01 cub. ft., and $60 \times 144 \times 7{\cdot}01 = 60{,}566{\cdot}4$ foot lbs. per lb.

To find the weight of steam required per horse-power per hour: Divide work done per horse-power per hour by work done per lb. of steam.

The work done per horse-power per hour $= 33{,}000 \times 60 = 1{,}980{,}000$ ft. lbs. The work done per lb. of steam at 100 lbs. pressure absolute without expansion $= 62{,}352$ ft. lbs.

Therefore, the number of pounds of steam required per horse-power per hour under the above conditions

$$= \frac{1{,}980{,}000}{62{,}352} = 31{\cdot}7 \text{ lbs.}$$

HEAT REJECTED BY STEAM TO CONDENSER

When steam is condensed, the heat rejected by it to the condensing water is not always the same, but depends upon the conditions under which it is condensed. If it is condensed under the same constant pressure at which it was formed, the heat given out will be the same as the total heat supplied; in other words, the heat rejected is the same as its total heat of formation; but if it be condensed under any other conditions, the heat rejected by the steam to the condensing water will be different. This statement may be illustrated by taking three cases:

1st case.—Referring again to fig. 15, stage 4, suppose that when the last particle of water is evaporated, we now commence to cool down the cylinder till the steam is condensed, and converted finally to water at 32°, the piston having fallen to its first position. Now, it will be evident that just as the formation of steam took place under the constant pressure of the weighted piston, so condensation has here been carried on under the same constant pressure, and the whole of the process of formation has been exactly reversed.

Hence, heat rejected by water in falling from 212° to 32° = 180 units; heat rejected by steam = heat absorbed in internal work = 893·7 units; and, lastly, heat expended in raising piston which has been restored to steam by piston compressing it back to original volume as water = 72·3 units; and, therefore,

$$\begin{aligned}\text{Heat rejected} &= 180 + 893\cdot7 + 72\cdot3 \\ &= 1{,}146 \text{ units.} \\ &= \text{total heat supplied.}\end{aligned}$$



FIG. 19.

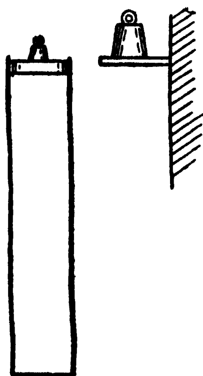
2nd case.—Suppose, in fig. 19, that, when the cooling commenced, the piston had been secured so that it could not fall as the volume of the steam decreased. Then evidently the heat rejected would be less than in the previous case by the amount of work done on the steam by the falling piston under atmospheric pressure; or,

$$\begin{aligned}\text{Heat rejected} &= \text{total heat} - \text{external work}\\ &= 1{,}146 - 72{\cdot}3\\ &= 1{,}073{\cdot}7 \text{ units}\end{aligned}$$

for this particular case.

This corresponds to the amount of heat rejected when the steam is exhausted to a condenser without back pressure.

3rd case.—Suppose now that the steam is exhausted into a condenser against a back pressure of say one-third of the pressure of the atmosphere. Then the effect is the same as though, when the piston had arrived at the extreme height due to the volume of 1 lb. of steam at 212° under the pressure of the atmosphere, the piston is secured, the weight representing the atmospheric pressure slipped off, and a weight one-third this size placed on the piston (fig. 20). Then, when the steam has been cooled till it only exerts a pressure of 5 lbs. per square inch, the piston will begin to fall, and, on continuing the cooling operation, the steam is condensed to water, and the water falls to 32°. Here the stages during the formation of steam have been reversed, except that the work done on the steam by the falling piston will be only $\frac{1}{3}$ of that done on the piston by the steam; hence


FIG. 20.

$$\begin{aligned}\text{Heat rejected} &= \text{heat of water from } 212° \text{ to } 32° = 180\\ &+ \text{internal heat of steam} = 893{\cdot}7\\ &+ \tfrac{1}{3} \text{ external work} = \tfrac{1}{3} \text{ of } 72{\cdot}3 = 24{\cdot}1\\ &\qquad 1{,}097{\cdot}8\end{aligned}$$

We shall now be able more fully to appreciate the meaning of the following definitions :

Sensible heat is the heat added to the water which changes its temperature, and the term is used to denote the heat required to raise the temperature of 1 lb. of water from 32° to the given temperature. Thus for water at boiling temperature under atmospheric pressure the sensible heat $= 212 - 32 = 180$.

If the temperature of the water to begin with is, say, 50° F.

instead of 32°, then the number of thermal units required to raise water at 50° to water at 212° = 212 − 50 = 162.

The latent heat of steam is defined as the amount of heat required to convert 1 lb. of water at a given temperature into steam at the same temperature.

The *total heat of evaporation* is the sum of the latent and sensible heats and is defined as the quantity of heat required to raise 1 lb. of water from 32° to the temperature of evaporation, and to convert it into steam at that temperature.

The total heat of evaporation for steam at any particular temperature, t, may be obtained approximately from the following formula:

$$\text{Total heat} = 1{,}082 + {\cdot}3\, t.$$

The latent heat may be obtained by subtracting $t - 32$ from the total heat found as above; or from the following formula:

$$\text{Latent heat} = 1{,}114 - {\cdot}7\, t.$$

Example.—Find the latent heat of steam at 120 lbs. pressure absolute, given that the temperature of steam at this pressure is 341° F.

$$\begin{aligned} \text{Then, latent heat} &= 1{,}114 - {\cdot}7\, t \\ &= 1{,}114 - {\cdot}7 \times 341 \\ &= 875{\cdot}3. \end{aligned}$$

The data required as to temperature, total heat, &c., of saturated steam are best obtained by reference to tables which have been prepared on the basis of the exhaustive experiments of Regnault. (See Table III.)

From the formulæ given above for the total and latent heats of steam, it will be evident that the total heat increases as the temperature of the steam increases, while the latent heat decreases as the temperature increases.

CHAPTER VII

SATURATED STEAM—TABLE OF PROPERTIES

STEAM in contact with the water from which it is generated is said to be saturated. It is then at its maximum density and pressure for the given temperature.

From the following table, p. 39, it will be seen that saturated steam under a given pressure has a fixed temperature, also that the temperature and density increase with the pressure. But it will be further noticed that the total heat increases in a very slow ratio compared with the pressure and temperature, there being only a very small increase of total heat per lb. of steam as the pressure increases. This is an important point in practice when considered in reference to coal consumption, for it shows that it is not much more costly in fuel to generate high-pressure steam than low-pressure steam, weight for weight; but we shall see further on that far more work can be obtained from high-pressure steam when used expansively than from the same weight of low-pressure steam, and hence the economy of high-pressure steam.

Example.—A cylinder contains 15 cub. ft. of steam at 40 lbs. absolute pressure: find the weight of this volume of the steam.

By Table III. steam at 40 lbs. absolute pressure occupies 10·28 cub. ft. per lb.

Then, 10·28 cub. ft. of steam at 40 lbs. pressure weigh 1 lb.

1	"	"	"	$\frac{1}{10{\cdot}28}$ lbs.
15	"	"	"	$15 \times \frac{1}{10{\cdot}28}$ lbs.
				$= 1{\cdot}46$ lbs.

III. *Table of Properties of Saturated Steam*

Absolute pressure in lbs. per sq. in.	Temperature Fah.	Total heat of evaporation from water at 32° F.	Volume per lb. in cubic feet.	Absolute pressure in lbs. per sq. in.	Temperature Fah.	Total heat of evaporation from water at 32° F.	Volume per lb. in cubic feet.	Absolute pressure in lbs. per sq. in.	Temperature Fah.	Total heat of evaporation from water at 32° F.	Volume per lb. in cubic feet.
1	102·0	1113·0	330·36	21	230·7	1152·3	18·84	85	316·1	1178·4	5·05
2	126·4	1120·5	172·08	22	233·3	1153·1	18·03	90	320·3	1179·6	4·79
3	141·6	1125·1	117·52	23	235·8	1153·9	17·26	95	324·1	1180·8	4·55
4	153·1	1128·6	89·62	24	238·2	1154·6	16·64	100	327·7	1181·9	4·33
5	162·3	1131·4	72·66	25	240·5	1155·3	16·00	105	331·3	1182·4	4·14
6	170·1	1133·8	61·21	26	242·7	1156·0	15·38	110	334·6	1184·0	3·97
7	176·9	1135·9	52·94	27	244·8	1156·6	14·86	115	338·0	1184·5	3·80
8	183·0	1137·7	46·70	28	246·8	1157·2	14·37	120	341·1	1186·0	3·65
9	188·4	1139·4	41·80	29	248·7	1157·8	13·90	130	347·[illegible]	1187·9	3·38
10	193·3	1140·9	37·84	30	250·5	1158·3	13·46	140	35[illegible]·9	1189·6	3·16
11	197·8	1142·3	34·63	35	259·3	1161·0	11·65	150	358·3	1191·2	2·96
12	202·0	1143·5	31·90	40	267·0	1163·4	10·28	160	363·4	1192·8	2·79
13	205·9	1144·7	29·57	45	274·4	1165·6	9·18	170	36[illegible]·3	1194·3	2·63
14·7	212·0	1146·6	26·36	50	281·0	1167·6	8·31	180	373·0	1195·7	2·49
15	213·1	1146·9	25·85	55	287·1	1170·0	7·61	190	377·5	1197·1	2·37
16	216·3	1147·9	24·32	60	292·6	1171·2	7·01	200	381·8	1198·4	2·26
17	219·5	1148·9	22·96	65	298·0	1172·7	6·49	250	400·8	1204·2	1·83
18	222·5	1149·8	21·78	70	302·8	1174·3	6·07	300	417·1	1209·2	1·54
19	225·3	1150·6	20·70	75	307·5	1175·7	5·68	350	430·1	1212·2	1·33
20	228·0	1151·5	19·72	80	312·1	1177·1	5·35	400	445·0	1217·7	1·18

Water heated in a Closed Vessel

Let water at 32° be heated in a closed vessel, such as an ordinary steam boiler, containing space for the accumulation of steam, and let heat be gradually applied. Then the temperature of the water will gradually rise, and steam will be formed at once, and not only when some definite temperature is reached, as was the case with the movable piston.

As the heat is increased, the temperature, pressure, and density, or weight per cubic foot, of the steam increase indefinitely, so long as the strength of the boiler is not exceeded; and the relation between the temperature, pressure, and density always bears a certain fixed relation, as given by Regnault's Tables, p. 39.

If heat is applied so as to maintain the temperature constant, the pressure and density remain constant also, and evaporation ceases. If a communication be opened between the boiler and engine, on escape of steam from the boiler the pressure is momentarily reduced and re-evaporation commences rapidly. So long as the temperature is maintained, no sensible variation of pressure is noticeable in a boiler supplying steam to an engine.

Temperature of Mixtures—Condensing Water

Example 1.—If 1 lb. of water at 212° F. be mixed with 5 lbs. of water at 50° F., find the temperature of the mixture.

Note.—In order to avoid confusion in problems of this kind, it is necessary to remember that the *total heat* in water or steam is always reckoned from 32° F. or 0° C. Hence it is necessary to subtract 32 from the temperature given in Fahrenheit degrees.

Let t = temperature required. Then

Total heat in 1 lb. of water at 212° + Total heat in 5 lbs. of water at 50° = Total heat in 6 lbs. of water at $t°$.

$$1(212-32)+5(50-32)=6(t-32)$$
$$180+90=6t-192$$
$$6t=462$$
$$t=77° \text{ F.}$$

Example 2.—How much water at 55° F. must be mixed with 1 lb. of *water* at 212° F. so that the resulting temperature of the mixture may be 105° F.?

Let W = weight of water required; then

Total heat in 1 lb. of water at 212°	+	Total heat in W lbs. of water at 55°	=	Total heat in (W + 1) lbs of the mixture at 105°
$1(212-32)$	+	$W(55-32)$	=	$(W+1)(105-32)$
180	+	$23W$	=	$73W+73$

$$50W = 107$$
$$W = 2 \cdot 14 \text{ lbs.}$$

In this connection it is interesting and important to compare the difference in the weight of water required to cool a given weight of *water*, with that required to cool the same weight of *steam* at the same temperature.

In the following example it is shown that it takes ten times as much water to cool 1 lb. of steam at 212° as it takes to cool the same weight of water at 212° to the same final temperature of 105°.

Example 3.—How much water at 55° F. will be necessary to condense 1 lb. of *steam* at 212° so that the resulting temperature in the vessel shall be 105° F., assuming condensation takes place at the pressure due to the temperature of the steam?

Let W = weight of water required; then

Total heat of 1 lb. of steam at 212°	+	Total heat in W lbs. of water at 55°	=	Total heat in (W + 1) lbs. of water at 105°
1146	+	$W(55-32)$	=	$(W+1)(105-32)$
1146	+	$23W$	=	$73W+73$

$$50W = 1073$$
$$W = 21 \cdot 46 \text{ lbs.}$$

Compare this answer with that in Ex. 2 above.

Example 4.—Find the temperature of the mixture when 21·5 lbs. of condensing water at 55° F. are used per lb. of steam at atmospheric pressure.

Let t = the temperature required; then

Total heat in 1 lb. of steam at 212°	+	Total heat in 21·5 lbs. of condensing water at 55°	=	Total heat in 22·5 lbs. of mixture.
1146	+	$21 \cdot 5(55-32)$	=	$22 \cdot 5(t-32)$
1146	+	$494 \cdot 5$	=	$22 \cdot 5\, t - 720$

$$22 \cdot 5\, t = 2360 \cdot 5$$
$$t = 104 \cdot 9° \text{ F.}$$

CHAPTER VIII

RELATION BETWEEN PRESSURE AND VOLUME OF GASES

Let a portion of gas be introduced into a cylinder which is closed at one end and fitted with a movable piston. Then the gas will fill every part of the space beneath the piston, and exert a uniform pressure on each square inch of surface with which it is in contact. If the internal volume of the cylinder be increased, by lifting the piston, the gas will still completely fill the space, but it will be less dense—that is, it will weigh less per cubic foot—and it will exert less pressure per square inch of surface with which it is in contact.

If the gas be compressed into a smaller space, it will become more dense, and it will exert a greater pressure per square inch.

The relation between the volume and pressure of a perfect gas at constant temperature is expressed in the following terms, known as Boyle's Law:

'The volume of a given portion of gas varies inversely as the pressure, the temperature remaining constant.'

This may be illustrated as follows:

Let a cylinder (fig. 21) be closed at one end and contain a movable piston, and let the piston, when in position *a*, enclose one cubic foot of gas under atmospheric pressure, or, say, 15 lbs. per square inch.

Suppose, now, that weights be added to the piston till the pressure on the enclosed gas is equal to 30 lbs. per square inch, or two atmospheres. Then, by the law just stated, the pressure

on the gas being doubled, the volume will be reduced one-half. Hence the piston now occupies position *b*, so that $e\,b = \frac{1}{2}\,e\,a$.

Again, apply to the piston a pressure equal to 60 lbs. per sq. in., or four atmospheres. The pressure on the gas being now four times the original pressure, its volume is one-fourth of its original volume, and the piston now falls to *c*, so that $e\,c = \frac{1}{4}\,e\,a$. Again, apply to the piston a pressure equal to 120 lbs. per sq. in., or eight atmospheres. The pressure on the gas being eight times the original pressure, the volume is now one-eighth of the original volume and the piston falls to *d*, so that $e\,d = \frac{1}{8}\,e\,a$. If now horizontals be drawn from the respective piston positions the length of which is equal to the pressure at these positions to any scale, and a curve be drawn through the extremities of the lines, the student will recognise the curve as being similar to that of an engine indicator diagram if the book be held so that the cylinder is horizontal. This curve is called a rectangular hyperbola.

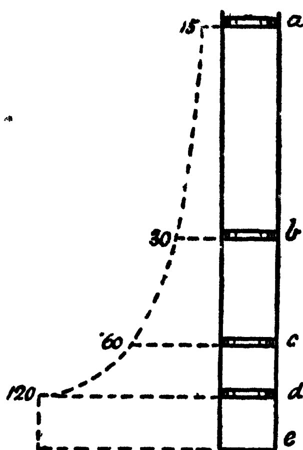


FIG. 21.

Boyle's Law may also be expressed thus :

If V is the volume	at	pressure	P
then $\frac{1}{2}$ V	„	„	2 P
$\frac{1}{3}$ V	„	„	3 P
$\frac{1}{4}$ V	„	„	4 P

and so on ; or,

2 V	„	„	$\frac{1}{2}$ P
3 V	„	„	$\frac{1}{3}$ P
4 V	„	„	$\frac{1}{4}$ P

From which it is evident that in each case, if the pressure be multiplied by the volume, the result is a constant number.

The Hyperbolic Curve

Boyle's Law and the properties of the hyperbolic curve may be further illustrated as follows: Suppose that one cubic foot of gas at 120 lbs. pressure is enclosed in a cylinder and expanded to eight times its original volume. Then the successive changes of volume and pressure may be represented by lines, thus: Draw two lines O M, O N from a common point O at right angles to one another. Let the vertical line O M be called the *line of pressures*, and the horizontal line O N the *line of volumes*.

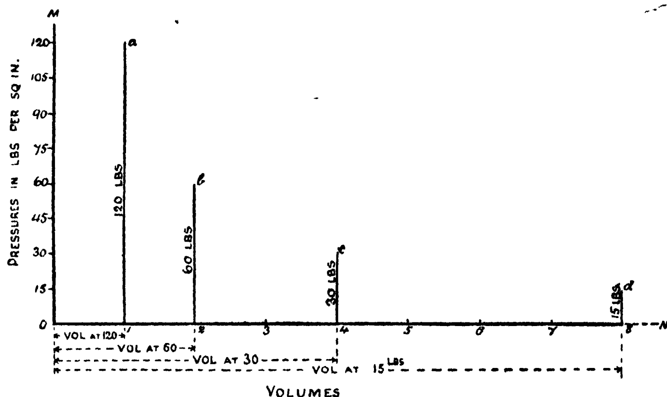


Fig. 22.

Mark off on the line of pressures to a scale of, say, $\frac{3}{16}$ inch = 15 lbs., a series of divisions, and on this scale place the figures 15, 30, 60, 120, &c., opposite the points representing these pressures.

On the line of volumes take, say, $\frac{3}{8}$ inch = 1 cub. ft., and mark 1, 2, 4, 8, &c., opposite their respective positions. From these points raise verticals to represent to scale the pressure of the gas at the various volumes. Thus $1\,a = 120 =$ the initial pressure of the 1 cub. ft. of gas. The gas is now expanded to 2 cub ft., the temperature meanwhile being supposed to be kept constant; and the pressure $2\,b$ will now have fallen to $\frac{1}{2}$ of $120 = 60$; at 4 cub. ft. the pressure $4\,c = \frac{1}{4}$ of $120 = 30$;

and at 8 cub. ft. the pressure $8d = \frac{1}{8}$ of 120 = 15. If the free ends *a*, *b*, *c*, *d* (fig. 22) of the verticals are now joined, the curve formed is called a rectangular hyperbola, fig. 23.

Then the four rectangles O*a*, O*b*, O*c*, O*d* represent the conditions of the gas as to volume and pressure for the respective piston positions, and these rectangles are all equal in area; for, $120 \times 1 = 60 \times 2 = 30 \times 4 = 15 \times 8 = 120$. In other words, pressure × volume = a constant, namely, in the present example, 120.

This curve represents the relative changes in volume and pressure for a perfect gas when the temperature is kept the

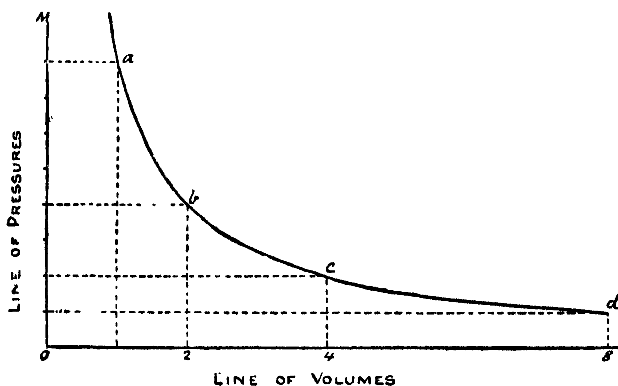


FIG. 23.

same throughout, hence it is called an *isothermal* curve, meaning the curve formed when the gas expands at *equal or uniform temperature*. The curve also describes fairly accurately the relation between the varying pressures and volumes of expanding steam in an engine cylinder. It is not, however, an 'isothermal' *for steam*, because the *temperature* of saturated steam varies with the pressure (see Table III., p. 39), unless it be superheated, which is not usually the condition of steam in a steam-engine cylinder.

Example.—Steam at 85 lbs. boiler pressure, or 100 lbs. pressure per square inch absolute, is admitted to a cylinder 5 ft. long, and cut off at $\frac{1}{5}$

of the stroke. Draw the theoretical indicator diagram, assuming that the hyberbolic curve is sufficiently accurate.

NOTE.—In drawing theoretical indicator diagrams, always use *absolute* pressures.

Let O M = line of pressures, and on it mark a scale of pressures, say $\frac{1}{10}$ inch = 5 lbs. Let O N = line of volumes to scale of, say, $\frac{5}{8}$ inch = 1 ft. of stroke of piston, and divide this line into ten equal parts. Complete the rectangle O M *a* A. Then O A = the volume of the steam and A *a* the pressure, at the point where the steam is cut off. To find the pressures B *b*, C *c*, &c., at B, C, D, &c., corresponding to the successive volumes O B, O C, O D, &c., advancing by distances along O N of 0·5 ft. Since the pressure at any point is inversely as the volume :

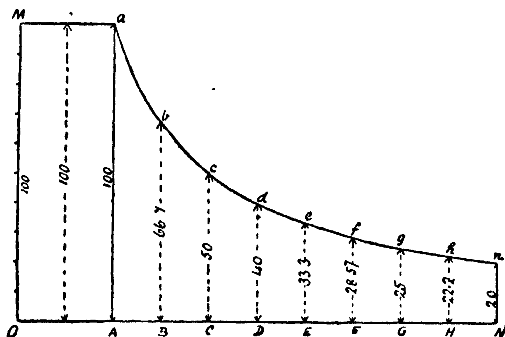


FIG. 24.

Pressure at $B = \frac{OA}{OB} \times \text{initial pressure} = \frac{2}{3} \times 100 = 66{\cdot}66$

" $C = \frac{OA}{OC}$ " " $= \frac{2}{4} \times 100 = 50{\cdot}00$

" $D = \frac{OA}{OD}$ " " $= \frac{2}{5} \times 100 = 40{\cdot}00$

" $E = \frac{OA}{OE}$ " " $= \frac{2}{6} \times 100 = 33{\cdot}33$

" $F = \frac{OA}{OF}$ " " $= \frac{2}{7} \times 100 = 28{\cdot}57$

" $G = \frac{OA}{OG}$ " " $= \frac{2}{8} \times 100 = 25{\cdot}00$

" $H = \frac{OA}{OH}$ " " $= \frac{2}{9} \times 100 = 22{\cdot}22$

" $N = \frac{OA}{ON}$ " " $= \frac{2}{10} \times 100 = 20{\cdot}00$

The above method of finding the pressure at any point during the expansion when the initial pressure is given may be expressed as follows: Multiply the initial pressure in lbs. per sq. in. by the length of stroke to point of cut-off, and divide by the distance of the given point from the beginning of the stroke.

The hyperbolic curve may be described without any calculation by the following simple geometrical method.

Draw the lines O M and O N as before. (NOTE.—The point O in the line O M is the zero of pressure, and not the point through which the line of atmospheric pressure passes.) Complete the parallelogram O M *a* A as in fig. 25. Produce M *a* parallel to O N. To find the pressure at any point B

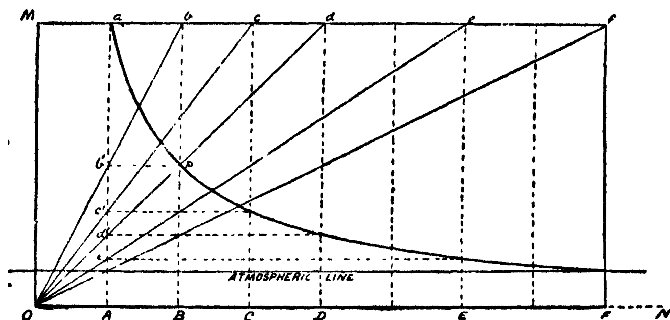


FIG. 25.

corresponding to the volume O B, draw the vertical B *b* and join O *b*, cutting A *a* in *b'*. Then the horizontal through *b'* to cut the vertical B *b* gives a point *p* in the curve. Any number of other points may be obtained in the same way, and the curve drawn through the points may be completed.

This curve describes the relation between the varying pressures and volumes of a gas, whether the gas is expanding or being compressed, provided the temperature remains constant.

Boyle's Law, however, is not absolutely obeyed by any known gas, and less so by steam; but the knowledge of the law is of great value in enabling us to obtain results which, for ordinary purposes, correspond with sufficient accuracy to the behaviour of the steam expanding in the cylinder.

CHAPTER IX

EXPANSIVE WORKING

WHEN engines are required to exert their full power for a short period—as happens, for example, with the locomotive in mounting an incline—steam is admitted to the cylinder at full pressure through the greater part of the stroke, without regard to economy in the consumption of steam or fuel. But this is not the way in which steam is used for any length of time in well-constructed and well-managed engines; and although extra work is obtained from the engine by neglecting to use the steam expansively, it is being very dearly paid for in the excessive proportion of steam and fuel consumed compared with the extra work done, as we shall now proceed to show.

WORK DONE BY STEAM USED EXPANSIVELY

We have seen that the work done per lb. of steam without expansion at high pressures only slightly exceeds that done by the same *weight* of steam at low pressures. We will now call attention to the increased work which may be obtained from high-pressure steam when advantage is taken of its expansive properties.

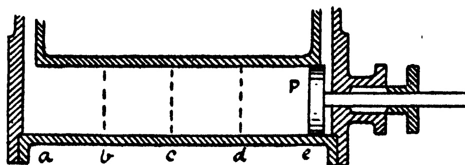


FIG. 26.

Let 1 lb. of steam at 100 lbs. per sq. in. absolute be admitted to a cylinder (fig. 26), when the piston P is at the end

a of the cylinder, and let the supply of steam be continued for one-fourth of the stroke, namely, till the piston reaches *b*, when we will suppose it just contains 1 lb. of steam. The supply is now cut off and the piston is driven for the remainder of the stroke by the expansive force of the steam thus enclosed. At the end of the stroke the steam occupies four times its original volume, and its pressure is now one-fourth its original or initial pressure, and the work done by the 1 lb. of steam will be clearly shown by the aid of a diagram. Thus let *a p* (fig. 27) be drawn to any convenient scale of pressures to equal 100 lbs.; and make $a\,v_1$ equal to 4·33 to any other convenient scale. (*Note*: 1 lb. of steam at 100 lbs. pressure absolute occupies 4·33 cub. ft., and if we assume the area of the piston =1 sq. ft., then length $a\,v_1=4{\cdot}33$ ft.) Produce the line to $a\,v_4$, making $a\,v_4=4$ times $a\,v_1$. Complete the figure by the graphical method (p. 47). Now the whole work done by the steam is equal to the area of the figure $p\,a\,v_4\,f\,r$, and this whole area is made up of two parts, namely:

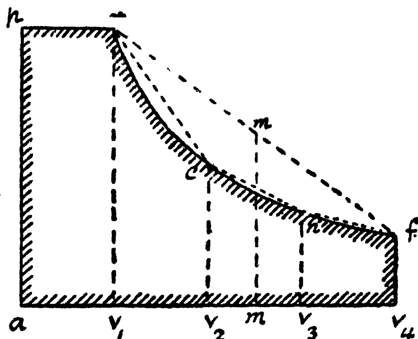


FIG. 27.

(1) area $p\,a\,v_1\,r$=work done during admission;
(2) area $r\,v_1\,v_4\,f$=work done during expansion.

Hence, by making use of the expansive properties of steam, we obtain the additional work out of it represented by the latter area.

To find the area of the figure would be a simple process if the line *r f* (fig. 27) had been a straight line instead of a curve, for then the area of the admission portion$=a\,p\times a\,v_1$; and the area of the expansion portion$=v_1v_4$ multiplied by the mean height *m m*. But the curve falls below this line, hence the

area thus obtained is too large, and the greater the expansion the greater the error. Much greater accuracy, however, is secured by this method if several divisions are taken, as shown by the dotted lines rc, cn, nf, in fig. 27, and the greater the number of divisions taken the greater the accuracy of the result. This is practically the method used by engineers in finding the area of the indicator diagram, the figure being divided into ten equal portions, as explained on p. 55.

The *exact* value of the expansion portion of a theoretical diagram may be readily obtained by referring to a table of hyperbolic logarithms; for, when the curve rf is hyperbolic, the hyperbolic logarithm expresses the relation between the area during expansion and the area during admission.

Thus, if the steam is cut off at half-stroke, it is expanded to twice its original volume; and if the area during admission = 1, then the area during expansion = the hyperbolic logarithm of 2; hence

total area = 1 + hyp. log. 2.

Now, on turning to a table (given in most engineers' pocket-books), the hyp. log. of 2 is ·696; then

total area = 1 + ·696
= 1·696;

and for a general case, if R = the ratio of expansion, or the volume of the steam at end of stroke divided by the volume at point of cut-off,

then area during expansion = hyp. log. R,
and total area = 1 + hyp. log. R.

Thus, in the example (fig. 27), the steam was cut off at one-fourth of the stroke, therefore at the end of the stroke the volume occupied by the steam was four times its volume at the point of cut-off; in which case R = 4, and the whole area = 1 + hyp. log. 4,

but by table (p. 51), hyp. log. of 4 = 1·386;
therefore whole area = 1 + 1·386 = 2·386.

That is to say, if the area $pav_1r = 1$, then the area $rv_1v_4f = 1{\cdot}386$, and the whole area = 2·386.

To express the work done in foot lbs. :

Work done during admission $= p\ v$

$= 100 \times 144 \times 4·33$

$= 62{,}352$ ft. lbs.

Total work done during admission and expansion to four volumes $= 62{,}352 \times 2·386$

$= 148{,}771·872$ ft. lbs.

Example.—Find the weight of steam required per indicated horse-power per hour, working at a pressure of 100 lbs. per sq. in. absolute, with a cut-off at one-fourth of the stroke, assuming there is no back pressure or loss from other causes.

$$\text{Then } \frac{\text{Work per I.H.P. per hour}}{\text{Work per lb. of steam}}$$

$$= \frac{1{,}980{,}000}{148{,}772} = 13·3 \text{ lbs.}$$

Compare this with result on p. 34.

The following table gives the proportional values of the work done for various degrees of expansion.

The work done by the steam during admission is taken as 1, and corresponds with the area $a\ p\ r\ v_1$ (fig. 27).

Steam in cylinder	Ratio of expansion R	Work done during admission	Work done during expansion = hyp. log. R	Total work done
Cut off at $\frac{3}{4}$ stroke	$1\frac{1}{3}$	1	0·262	1·262
,, $\frac{1}{2}$,,	2	1	0·693	1·693
,, $\frac{1}{3}$,,	3	1	1·098	2·098
,, $\frac{1}{4}$,,	4	1	1·386	2·386
,, $\frac{1}{5}$,,	5	1	1·609	2·609
,, $\frac{1}{8}$,,	8	1	2·079	3·079
,, $\frac{1}{9}$,,	9	1	2·197	3·197
,, $\frac{1}{10}$,,	10	1	2·302	3·302

From this table we see that, if steam is cut off at one-third of the stroke, and expanded to the end, the work done is about twice that done by the steam during admission.

(The exact proportion is 1 : 2·098.)

Now, if the steam had been admitted at initial pressure throughout the whole stroke, then three times the weight of steam would have been used, and the proportion of work then

done in the two cases, namely, supplying steam through the whole length of the stroke, or cutting off at one-third and expanding, would be as 3 : 2·098 ; in other words, to get half as much work again out of the engine, three times the weight of steam, and therefore also weight of fuel, is consumed in the first case as in the second.

The principle of the increased efficiency of steam with increased pressures and increased degrees of expansion may be further shown by the aid of the following diagram.

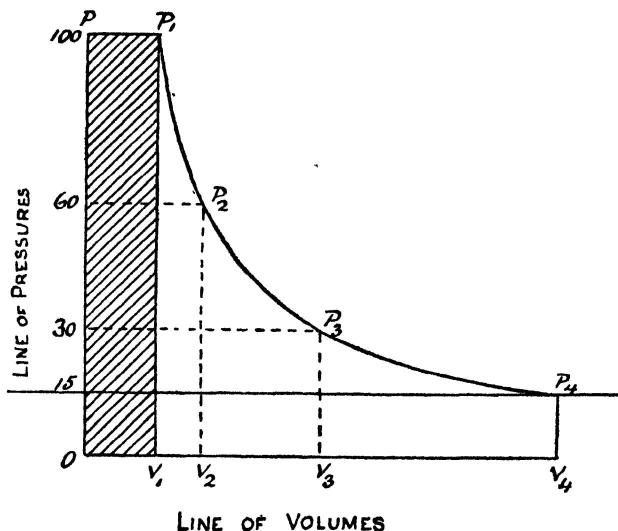


FIG. 28.

On the diagram (fig. 28) let oV_1, oV_2, &c., represent the volume occupied by 1 lb. of steam at pressures varying from 15 lbs. to 100 lbs. per sq. in. absolute.

Now, suppose in each case the steam is used non-expansively, that is, is supplied at full pressure throughout the stroke, and then allowed to escape into the air or condenser. Then, neglecting back pressure, the effect of which will be considered presently, the work done by the 1 lb. of steam under each of the several conditions is represented by an area as follows :

(1) Work done by steam at 100 lbs. = area o P_1 ;
(2) " " 60 lbs. = area o P_2 ;
(3) " " 30 lbs. = area o P_3 ,
(4) " " 15 lbs. = area o P_4.

But, assuming that steam obeys the law of Boyle, which is sufficiently accurate for our present purpose, these areas are all equal; hence the work done in each case is the same.

If now advantage be taken of the expansive power of steam, then with steam at 100 lbs. absolute, expanded down to 15 lbs., without back pressure, we are able to add to the area o P_1 the further area $P_1\,P_4\,V_4\,V_1$, which shows a very considerable increase in the work done. If the area o $P_1 = 1$, then the area $P_1\,P_4\,V_4\,V_1 = 1{\cdot}896$.

For the steam is expanded 6·66 times, hence area of whole

$$\begin{aligned} \text{figure} &= 1 + \text{hyp. log. } 6{\cdot}66 \\ &= 1 + 1{\cdot}896 \\ &= 2{\cdot}896. \end{aligned}$$

That is, if the work done by 1 lb. of steam at 15 lbs. pressure = 1, then by using the same weight of steam at 100 lbs. pressure and expanding down to 15 lbs. without back pressure, nearly three times the amount of work is done per pound of steam used, and practically also per pound of fuel consumed, for, as has been already shown, the consumption of fuel depends upon the *weight* of steam used, and is nearly independent of the *pressure* of the steam, owing to the fact that the total heat in steam at high pressures is only a very little greater than the total heat in steam of lower pressures (see table, p. 39).

Back Pressure

Back pressure has a considerable influence on the total work done by a given weight of steam.

Suppose the piston of a steam engine to be acted upon on one side by steam of 45 lbs. pressure absolute, and, if it be possible, let there be no pressure at all acting on the other side. Then, if the pressure of the steam were maintained uniform throughout the stroke, the diagram of pressures and volumes,

or, in other words, the diagram of work, would be a simple rectangle, thus (fig. 29):

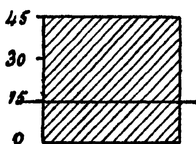


Fig. 29.

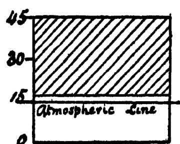


Fig. 30.

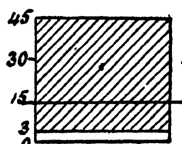


Fig. 31.

But in ordinary engines without a condenser, as the locomotive and most small factory engines, when the steam acts on one side of the piston, communication is open with the atmosphere through the exhaust passage on the other side, and it is therefore exposed to a back pressure of 15 lbs. per sq. in. (fig. 30). The effective pressure is therefore $45 - 15 = 30$ lbs. per sq. in.; and the effect on the diagram is to remove all the lower part from zero to 15 lbs., and thus reduce the area of the figure, and therefore also the effective work done. In practice there is an additional back pressure of 2 to 4 lbs., due to incompleteness of exhaust, making a total back pressure of 17 lbs. to 19 lbs. per sq. in. It may be much more than this with high-piston speeds. If, however, the cylinder were put, during exhaust, into communication with a condenser, then a large portion of the atmospheric pressure is removed, and a back pressure of not more than about 3 lbs. absolute will now oppose the motion of the piston. In this case the area of the figure representing the effective work done will be extended down to within about 3 lbs. of the zero line (fig. 31); the gain of work being proportional to the gain of area; while the weight of steam used in each case is clearly the same.

Effective pressure = Difference between pressures on each side of piston.

Mean Pressure

To find the mean effective pressure of steam per square inch on the piston, by measurement from the indicator diagram:

(1) Divide the line of volumes into ten equal parts.

(2) Measure the width of the figure at the centre of each division by the scale of pressures.

(3) Add the measurements together, and divide the sum by ten. The result gives the mean effective pressure per square inch on the piston.

To find the *total* mean pressure on the piston, multiply the mean pressure per square inch by the area of the piston in square inches.

Then the mean pressure on the piston in lbs., multiplied by the length of stroke in feet, gives the area of the figure, or the work done per stroke in foot lbs.

Example.—Find the mean effective pressure in the cylinder of a condensing steam engine when the pressure of steam on admission is 80 lbs. absolute, cut off at one-fourth of the stroke. Back pressure 3 lbs. per square inch.

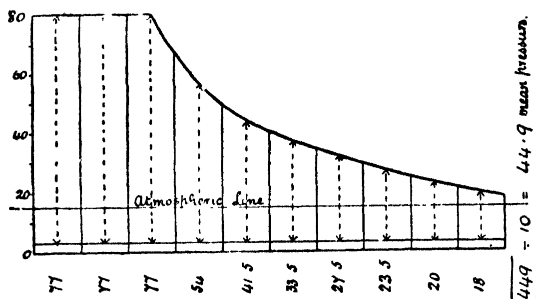


FIG. 32.

The same result might have been obtained for the theoretical diagram by using the following formula :

Let p = mean pressure of steam per sq in.
P = initial pressure, or pressure on admission to cylinder.
R = range of expansion, or ratio of volume at end of stroke to volume at point of cut-off.

$$\text{Then } p = P \times \frac{1 + \text{hyp. log. } R}{R} - \text{back pressure.}$$

Thus, for steam at 80 lbs. per sq. in. absolute, cut off at one-fourth of the stroke,

$$p = 80 \times \frac{1 + 1{\cdot}386}{4} - 3$$
$$= 47{\cdot}72 - 3$$
$$= 44{\cdot}72 \text{ lbs. per sq. in.}$$

The following is useful for reference in obtaining the theoretical mean pressure :

IV. *Table of Mean Pressures*

Number of times steam is expanded $= \frac{\text{final volume}}{\text{initial volume}}$	Mean pressure throughout stroke. Initial pressure = 1	Number of times steam is expanded $= \frac{\text{final volume}}{\text{initial volume}}$	Mean pressure throughout stroke. Initial pressure = 1
$1\frac{1}{3}$	·964	6	·465
$1\frac{1}{2}$	·937	7	·421
2	·846	8	·385
3	·699	9	·355
4	·596	10	·330
5	·522		

For back pressure, subtract from result obtained by above table, 3 lbs. for condensing engines and 17 lbs. for non-condensing engines, and the remainder will give the mean *effective* pressure.

Example.—Steam at 100 lbs. absolute is expanded down to 20 lbs., back pressure 17 lbs. ; find the mean effective pressure.

Here $\frac{100}{20} = 5$ = number of expansions

mean pressure (by table) = ·522

100 × ·522 = 52·2 lbs. mean absolute pressure, or,

52·2 − 17 = 35·2 lbs. per sq. in. mean effective pressure.

Indicated Horse-power

The diagram representing the work done on the piston has been called an indicator diagram. From this diagram, having obtained the mean pressure, the work done per stroke may be found. The work done per minute = the work done per

stroke × number of strokes per minute. The result may be expressed in horse-power by dividing the work done per minute by 33,000. The horse-power obtained from the indicator diagram in this way is called the *Indicated Horse-power.* It represents the effective work done on the piston by the steam.

The formula for Indicated Horse-power (I.H.P.) may be written, so as to be easily remembered, as follows:

$$\text{I.H.P.} = \frac{\text{units of work done per minute}}{33{,}000} = \frac{\text{P L A N}}{33{,}000}$$

Where P = mean effective pressure in lbs. per sq. in. on piston.
A = area of piston in sq. ins.
= (diameter of cylinder in inches)$^2 \times \cdot 7854$
L = length of stroke in feet, or distance travelled by the piston from end to end of cylinder.
N = number of strokes per minute; or
= number of revolutions of engine × 2.

For *work* is always estimated by a force of so many pounds acting through so many feet, and (P × A) pounds pressure on piston, acting through (L × N) feet per minute passed through = work done on piston per minute in foot lbs., which, divided by 33,000 = work done expressed in horse-power.

This formula will repay for careful study. It shows that a given indicated horse-power can be obtained by a variety of conditions, providing that the product P × L × A × N remains constant. Thus P, the mean pressure, may be made up by high-pressure steam cut off at an early point in the stroke, or by low-pressure steam acting through the greater part of the stroke. If P is increased by substituting high-pressure steam for low pressure, then A, the area of the piston, may be less, which means that the engine may be made smaller. If N, the number of revolutions, be increased, then L, the length of the stroke, may be decreased.

As a matter of fact, this is what has taken place in the development of the steam engine, namely, increased steam

pressures and higher piston speeds, which has resulted in a smaller, and therefore cheaper, type of engine. The early engines using low-pressure steam and running at a comparatively small number of revolutions assumed the type of the massive beam engine. The modern engine develops the same power with high pressures and high-piston speeds, and its dimensions are therefore proportionally decreased.

In the equation $\text{I.H.P.} = \frac{P\,L\,A\,N}{33{,}000}$ we have five indefinite terms, any one of which may be found when values are substituted for the remaining terms.

Example 1.—Find the indicated horse-power of an engine with a cylinder 12 ins. diameter, length of stroke 18 ins., number of revolutions per minute 90, mean effective pressure per square inch on piston 40 lbs.

$$\text{Then I.H.P.} = \frac{P\,L\,A\,N}{33{,}000}$$
$$= \frac{(P \times A)\text{ lbs.} \times (L \times N)\text{ ft. per min.}}{33{,}000}$$
$$= \frac{(40 \times 12 \times 12 \times \cdot 7854)\text{ lbs.} \times (1 \cdot 5 \times 90 \times 2)\text{ ft. per min.}}{33{,}000}$$
$$= \frac{4{,}520\text{ lbs.} \times 270\text{ ft. per min.}}{33{,}000}$$
$$= 37 \text{ nearly.}$$

Example 2.—An engine is required to indicate 37 horse-power with a mean effective pressure on piston of 40 lbs. per sq. in., length of stroke 18 ins., number of revolutions per minute 90; find the diameter of the cylinder.

First find the *area* from the formula:

$$\text{I.H.P.} = \frac{P\,L\,A\,N}{33{,}000}$$
$$A = \frac{33{,}000\text{ I.H.P.}}{P \times L \times N}$$
$$= \frac{33{,}000 \times 37}{40 \times 1 \cdot 5 \times 90 \times 2}$$

A, or area of piston = 113 sq. ins.

From which the diameter may be obtained thus:

$$\text{Diameter} = \sqrt{\frac{\text{Area}}{\cdot 7854}} = \sqrt{\frac{113}{\cdot 7854}} = \sqrt{144} = 12 \text{ inches.}$$

The horse-power of a compound engine is obtained in practice by finding the horse-power exerted in each cylinder separately from the indicator diagrams by the method above

explained, and adding the results together; the sum then gives the total indicated horse-power. Or, its theoretical value may be obtained from an ideal diagram, by considering that the whole of the work is done in the low-pressure cylinder only, working with steam at the initial pressure of the high-pressure cylinder, expanding down to the terminal pressure of the low-pressure cylinder.

EXAMPLES ILLUSTRATING ECONOMY OF EXPANSIVE WORKING

Example 1.—A condensing engine works with steam at 30 lbs. boiler pressure, cut off in the cylinder at half-stroke. It is proposed to increase the boiler pressure to 60 lbs. and to cut off at one-fourth of the stroke. Compare the relative work done and weight of steam used in the two cases.

—	Steam at 60 lbs. boiler pressure cut off at ¼ stroke	Steam at 30 lbs. boiler pressure cut off at ½ stroke
Absolute initial pressure . .	75 lbs.	45 lbs.
Theoretical mean pressure . .	44·7 lbs.	38 lbs.
Effective mean pressure, allowing 3 lbs. back pressure . .	41·7 lbs.	35 lbs.
Relative density or weight of steam per cub. ft.	75	45
Relative volumes used per stroke	1 vol.	2 vols.
Relative weight used per stroke	$75 \times 1 = 75$	$45 \times 2 = 90$

From which we gather that by using the higher pressure of steam with earlier cut-off there is a gain in effective mean pressure of $41·7 - 35 = 6·7$ lbs. per sq. in. $= \frac{6·7}{35} \times 100 = 19$ per cent., and a reduced consumption of steam $= 90 - 75 = 15$ lbs., saving on each 90 lbs. formerly used, or a saving of $\frac{15}{90} \times 100 = 16·6$ per cent.

As formerly explained, there will be a saving in fuel consumption corresponding with the saving in the weight of steam.

In practice it would be necessary to set against the above result—

(1) Probable increased initial condensation of steam in the cylinder, with the higher pressure and greater expansion.

(2) Increased initial stresses on the engine.

Example 2.—Two engines, single cylinder condensing, have cylinders of equal dimensions, each works with steam having a terminal pressure of 10 lbs. absolute, back pressure 4 lbs. The boiler pressure by gauge for the first engine is 60 lbs. and for the second 45 lbs. Compare the result in the two cases.

	1st case	2nd case
Boiler pressure	60 lbs	45 lbs.
Absolute initial pressure . .	75 lbs.	60 lbs.
Terminal pressure . .	10 bs.	10 lbs.
Cut-off (neglecting clearance)	$\frac{10}{75}$	$\frac{10}{60}$
Theoretical mean pressure .	30 lbs.	27·9 lb
Effective mean pressure, allowing 4 lbs. back pressure. .	26 lbs.	23·9 lb

The weight of steam used per stroke is the same in each case, for the cylinders are the same size, and the terminal pressures are equal. They, therefore, hold at the end of the stroke equal volumes of steam at the same pressure, and, therefore, of the same weight.

It is true that the total heat required to generate steam at 75 lbs. absolute is greater than that required for steam at 60 lbs. absolute, but this difference is so small that it may be neglected, and the consumption of fuel per lb. of steam generated may be assumed to be the same. There is, however, as seen by the table, a gain in mean effective pressure of $26-23·9=2·1$ lbs., or a gain of $\frac{2·1}{23·9}\times 100=9$ per cent. in power exerted with the higher pressure. To set against this we have greater initial stresses on the parts of the engine working with the higher pressure, in the proportion of $\frac{75-4}{60-4}=\frac{71}{56}$, or an increase of 27 per cent., requiring an engine in this case 27 per cent. stronger.

To further illustrate the advantage of expansive working,

we will take another case from actual practice. A steamer of 1,000 I.H.P., having a pair of two-cylinder compound oscillating paddle-wheel engines, made by Messrs. Laird of Birkenhead, runs at the following speeds and coal consumption for varying degrees of cut-off in each cylinder:

Point of cut off	Knots per hour	Coal Consumption per hour in cwts.	Coal Consumption per mile in cwts.
3-10ths	8	6	0·75
4-10ths	9	9	1·00
5-10ths	10	12	1·20
6-10ths	12	20	1·66

From this table it will be seen how the weight of steam supplied to the cylinder affects the speed and coal consumption. Comparing the effects of the two extreme points of cut-off, when cutting-off at 6-10ths instead of 3-10ths, twice the volume and weight of steam is used, 3·3 times the weight of coal is consumed per hour, and 2·2 times the weight of coal is consumed per mile, for 1·5 times the number of revolutions.

The effect of expansive working on the possible distance which a vessel can run with a given weight of fuel will also be evident; for in the case we are considering the vessel would run 2·2 times the distance when cutting-off at 3-10ths that she would run when cutting-off at 6-10ths. The influence of this increased economy of expansive working on the power to run longer voyages where coal is not easily obtained has had immense influence on British commerce with distant parts of the globe.

Limit of useful Expansion of Steam

The amount of back pressure in a cylinder determines the *limit of useful expansion* of the steam in the cylinder, for it would evidently be useless to expand the steam to a pressure below that of the pressure at the back of the piston.

Thus, let steam at 45 lbs. per sq. in. boiler pressure, or 60 lbs. per sq. in. absolute, be admitted to a cylinder, and cut off at one-fourth of the stroke of the piston. Let the engine be non-condensing with a back pressure of 15 lbs. per sq. in.

Then, referring to fig. 33, the useful work done by the expanding steam is represented by the shaded portion from the commencement of the stroke to the point *d* on the line of

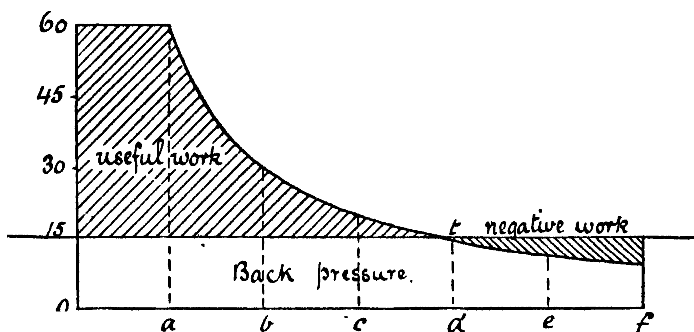


Fig. 33.

volumes. Here the pressure of the steam ($d\,t$) and the back pressure are equal, and it will be evident that any further extension of the diagram would be useless. If the expansion be continued beyond four, to five or six expansions, the work done by the atmosphere against the piston in the later stages is greater than that done on it by the steam, to the extent represented by the area of the shaded part marked *negative work.*

Hence, in an engine cylinder the steam should never be cut off so early as to cause it to expand to a pressure below that of the back pressure acting against the piston. In practice the expansion cannot be carried with advantage so far as this. Theoretical limit of number of expansions $=\frac{\text{initial pressure}}{\text{back pressure}}$.

Thus, in above case $\frac{\text{initial pressure}}{\text{back pressure}}=\frac{60}{15}=4$ expansions. In practice the maximum number of expansions should not exceed three-fourths of the theoretical limit. In condensing engines the steam is expanded down to a final pressure of about 10 lbs. per sq. in. absolute. And by dividing the known initial absolute pressure by the known terminal pressure we determine the number of expansions required. Thus, for a condensing engine

working with steam at an initial pressure of 150 lbs. absolute and expanding to a terminal pressure of 10 lbs. absolute,

$$\text{number of expansions} = \frac{\text{initial pressure}}{\text{terminal pressure}} = \frac{150}{10} = 15.$$

Such a large number of expansions could not be economically carried out in one cylinder, but in practice would require three successive cylinders

Clearance in the Cylinder

When the piston in a cylinder is at the end of its stroke it does not *touch* the end or cover of the cylinder, but there is always a certain space left between them to prevent the danger of their coming into actual contact. In addition to this is the passage between the face of the slide valve and the cylinder by which the steam is conducted to the cylinder. These two spaces (marked *c c*, fig. 34), which make up the whole space between the face of the valve and the face of the piston when the piston is at the end of its stroke, are called the *clearance*.

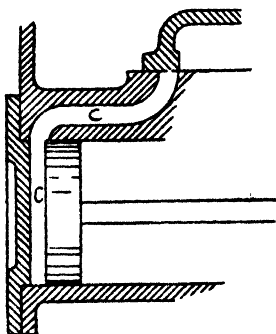


Fig. 34.

Let the volume displaced by the piston during its stroke = 9 cub. ft.; and the volume of the clearance = 1 cub. ft. Then, when steam is admitted, 1 cub. ft. is used to fill the clearance space before the piston moves; and if steam is used at full pressure throughout the stroke, 9 cub. ft. more is required to displace the piston. Thus 1 cub. ft. in every 10 cub. ft. of steam passes through the engine without doing any work, representing a loss of 10 per cent.

But suppose the steam is cut off at $\frac{1}{9}$th of the stroke. Then, during admission there is first 1 cub. ft. of steam to fill the clearance, and which so far does no work on the piston, and then 1 cub. ft. to displace the piston, when the steam is cut off. There are now 2 cub. ft. of steam at initial pressure

enclosed in the cylinder. Expansion commences, and at the end of the stroke the volume occupied by the steam will evidently be 10 cub. ft. Hence, pressure of steam at end of stroke $=\frac{2}{10}$ths, or $\frac{1}{5}$th of initial pressure.

If there had been *no* clearance, then we should have had 1 cub. ft. of steam in the cylinder at point of cut-off, which would expand to 9 cub. ft. with a terminal pressure of $\frac{1}{9}$th, the initial pressure.

Suppose the initial pressure had been 180 lbs. absolute. Then, neglecting the effect of clearance, the terminal pressure $=\frac{180}{9}=20$ lbs. per sq. in. absolute.

Including the effect of clearance, the terminal pressure $=\frac{180}{5}=36$ lbs. per sq. in. absolute, or nearly twice the terminal pressure obtained neglecting clearance.

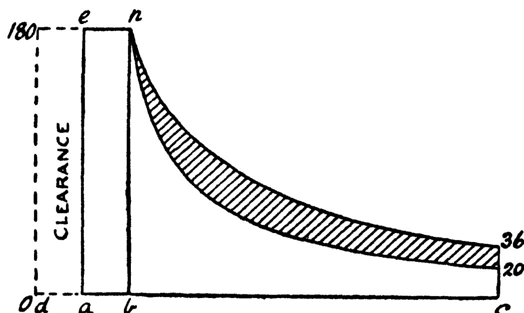


FIG. 35.

Although the steam required to fill the clearance space does no work on the piston during admission, yet when cut-off takes place the piston receives the advantage of the expansive force of this steam, and its effect in increasing the total work done is shown by the shaded part of the diagram (fig. 35).

To draw the diagram, set off *a c*=the stroke of the piston, to any scale and divide it into nine equal parts, construct the curve *p*, 20, by the graphical method from the point *a*, representing the expansion of steam of volume *a b* and pressure *b p*. To the

left of a draw $a\,d$, making $a\,d=\frac{1}{8}$th $a\,c$, that being the proportion of the volume of the cylinder occupied by the clearance space. Draw the curve p, 36, from the point d, representing the expansion of steam of volume $d\,b$, and pressure $d\,p$.

The loss by clearance may be much reduced by closing the exhaust passage in the cylinder before the end of the stroke, so that the steam so enclosed may be compressed and fill the clearance space at a pressure and temperature approaching that of the newly entering steam.

Priming

A boiler is said to 'prime' when the steam supplied by it to the cylinder is not dry, but is charged with more or less moisture.

Priming is frequently due to rapid evaporation, too small steam space, defective circulation, or the presence of certain impurities in the water. Of the total water used by a boiler, from 5 to 10 per cent. may be accounted for by priming.

Cylinder Condensation

When steam is admitted to a cylinder which is colder than the entering steam, the steam parts with some of its heat to the cylinder walls, a portion of the steam is condensed and deposited on the metallic surface, and more steam from the boiler enters the cylinder to take its place, while the temperature of the cylinder rises to that of the steam in contact with it.

If the steam be supplied to the cylinder at the initial pressure and temperature throughout the whole stroke, and the exhaust port be then opened, the steam will escape into the air, and the pressure in the cylinder will fall to that of the atmosphere, or nearly so.

But the water (which exists probably as a film), being in contact with the metallic walls of the cylinder at the temperature of the initial steam, will evaporate immediately the pressure is reduced by the opening of the exhaust, and become reconverted into steam at the expense of the heat in the walls of the

cylinder, thereby cooling them to the temperature of the steam during exhaust.

The steam thus re-evaporated during exhaust not only absorbs heat, which will have to be made up again from the entering steam during the next stroke, but it passes away to the air without doing any useful work; in fact, it acts rather as back pressure against the piston.

When the piston reaches the end of its stroke, the boiler steam is readmitted into the cylinder and comes in contact with the cooled surface of the cylinder cover, piston, and steam passages, which have been exposed to the temperature of the exhaust steam, and the same process of condensation and re-evaporation will be repeated.

If the cylinder had been in communication with a condenser instead of with the air, the temperature of the cylinder during exhaust would have fallen still lower, namely, to that due to the decreased pressure in the condenser, and the condensation of the initial steam during admission would have been still greater.

If the steam is cut off at an early point in the stroke, condensation occurs, as before, during admission, while the steam is hotter than the cylinder; but as the expansion proceeds, the pressure of the steam is reduced below that at which water at the increased temperature of the cylinder evaporates, and a portion of the condensed steam is consequently re-evaporated, the re-evaporation increasing as the pressure decreases towards the end of the stroke. On the opening of the port to exhaust, the pressure is still further reduced, and re-evaporation is completed.

Condensation, then, takes place during the early part of the stroke, while re-evaporation occurs partly towards the end of the stroke and partly during exhaust. The re-evaporation during expansion behind the piston helps the piston, and increases the total work done; but the steam re-evaporated during exhaust in a single cylinder engine passes away to waste. The loss due to condensation of steam in the cylinders of all engines varies from 10 to 50 per cent., or more, of the whole steam consumed, the loss becoming greater as the degree of

expansion increases, or, in other words, as the variation of temperature in the walls of the cylinder increases.

The economical advantage of using high-pressure steam is due to the power it possesses of doing work by expanding behind the piston after the supply is cut off from the boiler.

But the temperature of saturated steam varies with the pressure, and, therefore, if, in a single cylinder, steam at high pressure and temperature be admitted and expanded to a low pressure and temperature, the greater the degree of expansion the greater the difference in temperature between the steam on entering and leaving the cylinder.

Thus, suppose in each of the following cases the temperature of the exhaust to be that at atmospheric pressure, namely, 212° F. If the initial pressure of the steam—that is, the pressure on admission to the cylinder—be 45 lbs. absolute, temperature 274° F., we have a difference of temperature in the cylinder of $274-212=62°$ F., and on the admission of steam for the new stroke we shall have steam at 274° coming in contact with cylinder walls at 212°.

But suppose the initial pressure of the steam is raised to 90 lbs. absolute, temperature 320° F., which is expanded in the cylinder and exhausted into the atmosphere, then the difference of temperature in the cylinder is $320-212=108°$ F., and steam on admission at 320° comes into contact with cylinder walls at 212°.

But the loss due to initial condensation of steam in the cylinder increases as the variation in temperature in the walls of the cylinder increases; hence there is a limit to the useful expansion of steam in a single cylinder, owing to the excessive condensation in the cylinder, with high degrees of expansion, resulting in increased consumption of fuel instead of a saving, and giving rise to the expression 'Expansive working is expensive working.'

The secret of economy is in the carrying out of the principle laid down by Watt, namely, that *the cylinder should be kept as hot as the steam which enters it*, and engineers from Watt's day to the present have striven to accomplish this result.

The laws which govern the condensation of steam in the

cylinder are not at present well understood. The remedies so far adopted, which are, however, only partially successful, are :

(1) *Jacketing the cylinder* with hot steam (an example of jacketed cylinders is given in figs. 105 and 106). The addition of the steam jacket has a considerable influence in reducing the amount of condensation in the cylinder. The jacket is the more necessary the greater the degree of expansion in one cylinder, and the slower the piston speed.

(2) *Cushioning*, that is, compressing a portion of the exhaust steam by closing the exhaust port before the end of the stroke, and allowing the piston to compress the steam and thereby raise its pressure and temperature, and therefore also the temperature of the cylinder cover, steam passage, and piston, before the new steam is admitted.

(3) *Compounding the cylinders*, that is, adding one or more separate cylinders into which the steam may be expanded, and thereby reducing the variation of temperature in each cylinder.

CHAPTER X

THE STEAM ENGINE

Non-condensing Engines

Engines which exhaust their steam into the air after it has done work in the cylinder are called non-condensing engines.

The locomotive and most factory and mill engines belong to this type. Non condensing engines are known by the puffing of the escaping steam up the chimney, a phenomenon which is familiar to every reader. The puff of the exhaust steam occurs as the piston arrives at the end of its stroke, the escaping steam having previously driven the piston from one end of the cylinder to the other.

In condensing engines there is no puffing of exhaust steam into the air, the steam being passed instead into a box or condenser, where it is cooled and condensed by actual contact with a jet of cold water, or by contact with cold pipes through which cold water is flowing.

The essential parts of all ordinary non-condensing engines, whether the engine be an horizontal or vertical one, are practically the same, the difference in appearance among engines by different makers being due for the most part to a difference in shape or arrangement of the essential details.

The following diagrams (figs. 36 and 37) give a front view and side view of a small vertical non-condensing steam engine, as in use for various kinds of factory and mill work. The pressure of steam used in such engines is about 60 lbs. per sq. in. above the atmosphere.

The action of the parts is as follows: The steam is con-

ducted from the boiler by the steam pipe to the slide jacket or chamber in which the slide valve S V works. Here, by a sliding motion of the slide valve on the face of the ports, the

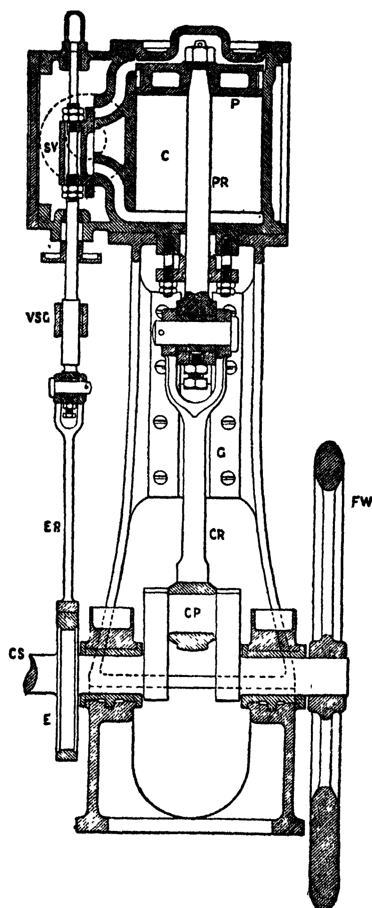


FIG. 36.

C, cylinder; P, piston; P R, piston rod; G, guides; C R, connecting rod; C P, crank pin; E, eccentric; E R, eccentric rod; S V, slide valve.

steam passages or ports are alternately opened, admitting steam to one side of the piston, and allowing it to escape from the other side into the air ; or, if a condensing engine, into a con-

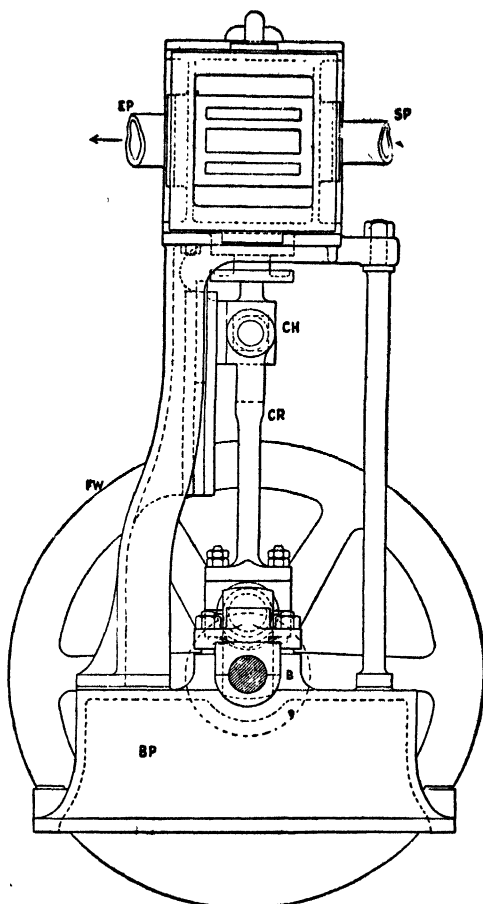


Fig. 37.

CH cross-head ; CR connecting rod ; B, bearings ; EP, exhaust pipe ; SP, steam pipe.

denser. (The exact action of the slide valve will be explained more fully presently.) The piston is thus made to move from end to end of the cylinder against the resistance due to the load which is communicated through the piston rod.

Attached to the outer end of the piston rod is the crosshead, having a flat base called a slipper, which slides to and fro between guides, and compels the piston rod to move parallel to the axis of the cylinder, thus preventing the angular action of the connecting rod from bending the piston rod. The connecting rod is attached at one end to the crosshead by a pin, sometimes called a gudgeon, which passes completely through the block and the fork end of the rod as shown, and at the other end to the crank pin. The reciprocating motion of the piston is by this means converted into the circular motion of the crank pin and shaft, and from the shaft by means of a pulley and belt, or by wheel gearing, the power of the engine is transmitted as required. See also figs. 104 and 105.

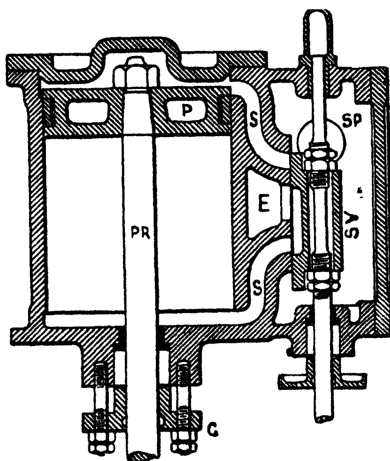


FIG. 38.
SP, steam pipe; S, steam port; E, exhaust port; SV, slide valve; P, piston, PR, piston rod; G, gland.

ENGINE DETAILS

The Cylinder.—The cylinder, which is made of cast-iron, consists of the cylindrical chamber, bored out perfectly true, and of the slide jacket or valve box. The cylindrical chamber is connected at each end with the slide jacket by passages called *steam ports*, S, through which steam passes to or from the cylinder. The passage between the two steam ports leads to the air, or to a condenser, and is called the *exhaust port*, E. This passage is put in

communication with either end of the cylinder as required by means of the slide valve. The ends of the cylinder are closed by covers bolted to the flanged ends. In the example (fig. 38) the bottom end is cast solid with the body of the cylinder.

In order to make the hole in the cover through which the piston rod passes steam-tight, a *stuffing box* is used, the construction of which will be understood from the figure. The casting is so formed as to leave a small space around the rod, which is filled with packing, or stuffing, consisting of tallowed hemp or other substitute, and the packing is pressed down on the rod by means of a cover or *gland* fitted with two screwed bolts. A similar arrangement of stuffing box and gland is fitted to the slide valve rod; it is also used for pump rods and other similar purposes.

The steam passages should be made as short as possible, because at each stroke the passage must be filled with its own volume of steam before the steam acts upon the piston. The effect of this has been described under the heading of *Clearance*, on p. 63.

The steam ports must be made large enough to admit sufficient steam to the cylinder during the instant the port is open, otherwise the steam will be *wiredrawn*.

Wiredrawing is the gradual fall of pressure of the steam behind the piston, as it proceeds on its stroke, owing to small and restricted steam passages. Its effect may be illustrated by the diagram (fig. 39). If the pressure of the steam on admission to the cylinder = O A, then the pressure, instead of being maintained at a pressure N E. to the point of cut-off, E, gradually falls from A to B.

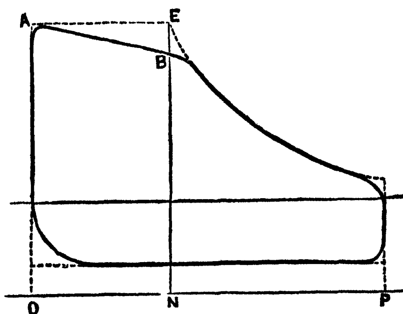


FIG. 39.

The stroke of the piston from end to end of the cylinder

(which is equal to the diameter of the crank-pin path) determines the internal length of the cylinder from cover to cover, which must evidently be equal to the stroke of the piston, plus the thickness of the piston, plus twice the clearance allowed between the piston and cylinder cover, when the piston is at the end of its stroke. This clearance, which is kept as small as possible, varies from $\frac{1}{8}$ in. to $\frac{1}{2}$ in., according to the size of the cylinder.

It will be noticed that the shape of the cylinder cover must be made to conform to that of the piston, otherwise a considerable volume of steam might be wasted at each stroke, in filling unnecessarily large clearance spaces.

Cylinder liner. Steam jacket.—Cylinders are sometimes fitted with a separate internal barrel, called a cylinder *liner*, as shown in the sectional view of the compound engines (fig. 105), made of hard cast-iron or of steel.

Between the liner and the body of the cylinder is a space called the *steam jacket*, which is filled with steam direct from the boiler. The depth of the jacket is about the same as the thickness of metal in the cylinder. Sometimes the cylinder covers are jacketed, as well as the body of the cylinder.

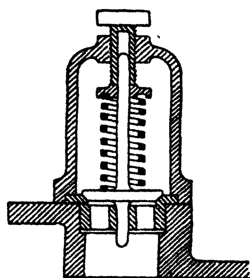


FIG. 40.

Cylinder escape valves.—To avoid the danger of the piston bursting the cylinder cover as it approaches the end of its stroke, owing to the occasional presence of water through priming or condensation, cylinder escape valves are often fitted on the cylinder covers.

The diagram (fig. 40) will explain the construction of these valves. The valve is of the ordinary conical kind, kept in position by a spring loaded a little above the pressure in the boiler.

Cylinder relief cocks (fig. 41) are also fitted to all cylinders to drain off the water, or to blow through the cylinder with the steam, and thus clear it of water, especially on starting the engine.

Example 1.—A cylinder is 15 ins. diameter, stroke of piston 25 ins.; find the capacity of the cylinder, allowing an addition of 7 per cent. for clearance space. *Ans.* 4726·7 cub. ins.

Note.—This represents the volume of steam in the cylinder at end of stroke; the following example shows how to find the weight of this volume.

Example 2.—Find the *weight* of 4726·7 cub. ins. of steam at 20 lbs. pressure per sq. in. absolute.

By Table III. 1 lb. of steam at 20 lbs. pressure absolute occupies 19·7 cub. ft.

Then 19·7 cub. ft. of steam at 20 lbs. pressure weigh 1 lb.

·1	,,	,,	,,	$\frac{1}{19 \cdot 7}$ lb.
$\frac{4726 \cdot 7}{1728}$	,,	,,	,,	$\frac{4726 \cdot 7}{1728} \times \frac{1}{19 \cdot 7}$ lbs.
				$= \cdot 1388$ lb.

The above two examples give the volume and weight of steam used per stroke in a cylinder of the above dimensions,

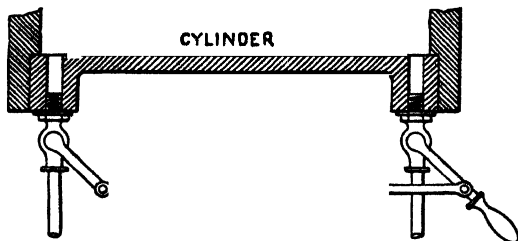


Fig. 41.

working with steam at 20 lbs. terminal pressure. To find the volume or weight of steam passing through the engine as steam vapour in a given time, multiply the above results by the number of times the cylinder is filled; in other words, multiply by the number of strokes made by the piston in the given time.

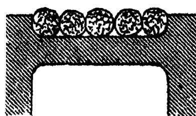
Example 3.—The engine in the above case runs at 100 revolutions per minute; find the weight of steam used per hour.

In 1 stroke the weight of steam used			$= \cdot 1388$ lb.
,, 1 revolution	,,	,,	$= (\cdot 1388 \times 2)$ lb.
,, 1 minute	,,	,,	$= (\cdot 1388 \times 2 \times 100)$ lbs.
,, 1 hour	,,	,,	$= (\cdot 1388 \times 2 \times 100 \times 60)$ lbs.
			$= 1665 \cdot 6$ lbs.

Example 4.—Suppose it is known that the horse-power of the above engine, when working at 100 revolutions, is 90; find the number of lbs. of steam used per horse-power per hour. *Ans.* 18·5

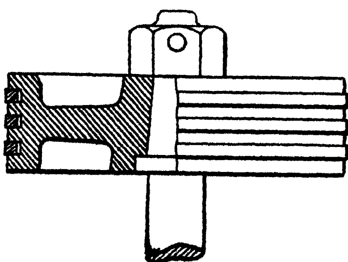
PISTONS

The piston is the movable plug which moves from end to end of the cylinder, under the pressure of the steam, and through which the energy of the steam is converted into the motion of the mechanism.


FIG. 42.

The piston must form a steam-tight division between the two ends of the cylinder. If it were possible to turn up a solid piston, which should so exactly fit the bore of the cylinder that it would be steam-tight, and at the same time move freely without friction, this would be a perfect piston.

In the early days of the steam engine, when steam pressures were very low, pistons were made steam-tight by coiling rope or *junk* in a groove on the rim of the piston, and this method is still adopted for pump buckets which only require to be water-tight. But for the pistons of steam cylinders a more perfect arrangement was soon found necessary. As at present made, the body of the piston is turned to an easy fit in the cylinder, and it is then made steam-tight by means of spring rings.


FIG. 43.

A common and simple arrangement is that of Ramsbottom's spring rings, which are simple steel or gun-metal rings of $\frac{1}{4}$ in. to $\frac{3}{8}$ in. square section (fig. 43). They are turned at first to a diameter a little larger than that of the cylinder they are required to fit; and a small piece is then taken out to enable them to close up to the bore of the cylinder when in their place. They are then sprung over the piston and fitted into grooves turned in the piston rim (fig. 43).

Figs. 43 and 44 are types of locomotive pistons; fig. 44 is fitted with two cast-iron packing rings about $\frac{1}{2}$ in. thick by $\frac{3}{4}$ in. wide, turned, cut and sprung into position as before. The rings are sometimes placed in the same groove and sometimes in separate grooves.

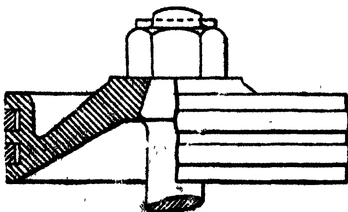


FIG. 44.

For marine work, pistons of the type shown in fig. 45 are much used. The packing ring consists of one large cast-iron ring, P R, which is pressed outwards against the cylinder by means of a series of

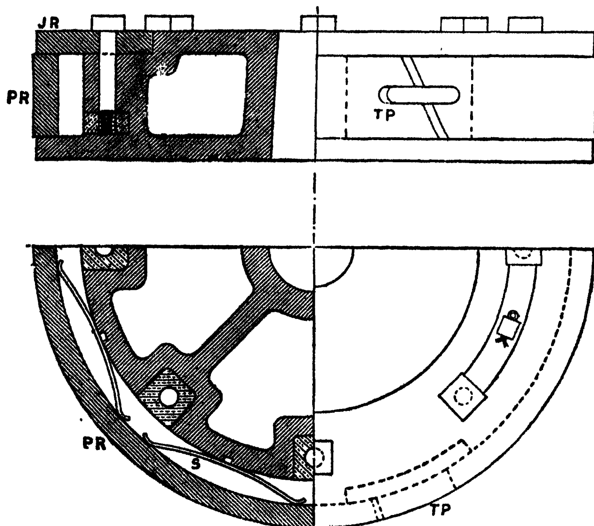


FIG. 45.

J R, junk ring; P R, packing ring; T P, tongue piece; S, spring.

springs, S, placed behind the packing ring. For horizontal cylinders, the bottom spring is removed and a cast-iron block is substituted, which takes the weight of the piston. Instead of

the small separate springs, various patent coiled springs are used in vertical engines.

The packing ring is turned a little larger (⅛ in. per foot diameter) than the bore of the cylinder ; it is then cut through by an oblique slit and tends to spring open as wear takes place. The steam is prevented from leaking through this opening by a brass tongue piece, T P, which is fitted in another groove cut across the slit as shown. The tongue piece is secured to a plate fastened to the back of the ring, and on one side of the slit (fig. 46).

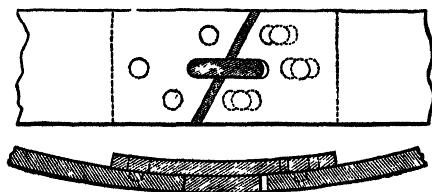


FIG. 46.

The packing ring is held in its place between two flanges, one of which is cast solid with the piston, the other being formed by a loose flat ring called the junk ring, J R. The junk ring is secured to the piston by screwed bolts which screw into brass nuts inserted in a cavity left for the purpose in the body of the piston. These bolts are prevented from slacking back by a guard ring or pieces of a ring fitted between the heads, as shown.

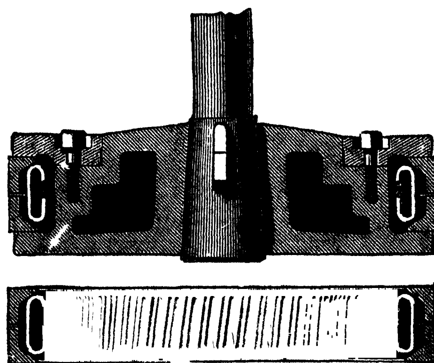


FIG. 47.

Fig. 47 is a section of Buckley's Patent Piston. The packing consists of two separate rings with a continuous coiled spring behind it. The action of the coiled spring is to keep the rings steam-tight, not only against the cylinder, but against the junk ring and flange of the piston.

The friction between the packing ring and the cylinder should be as little as possible consistent with steam-tightness, and the piston should be as light as possible consistent with strength. Steel pistons are now becoming common, and by using this material the weight of the piston can be considerably reduced.

A fruitful and all too common source of loss of efficiency in steam engines is the presence of leaky pistons, the steam passing from one side of the piston to the other. Such steam is worse than wasted, as it not only does no work on the piston but acts as back pressure against it. The pistons of locomotives are usually kept in good condition, and the short sharp exhaust of the locomotive is in striking contrast with the asthmatical exhaust of too many factory and mill engines.

Piston speed.—The mean speed of the piston in feet per minute = length of stroke × number of revolutions per minute × 2.

Example.—An engine with a 3-ft. stroke makes 80 revolutions per minute; find the mean speed of the piston.

$$3 \text{ ft.} \times (80 \times 2) \text{ strokes} = 480 \text{ ft. per minute.}$$

The mean speed of the piston in practice varies from about 250 ft. per minute for small stationary engines, to from 500 to 750 ft. per minute for marine engines, and in some cases it exceeds 1,000 ft. per minute in the locomotive.

There has been, and is, an increasing tendency towards high piston speeds and light moving parts.

Piston displacement per minute is the space swept through by the piston at each stroke, multiplied by the number of strokes per minute; = the area of the piston in square feet, multiplied by the speed of the piston in feet per minute.

Example.—Find the displacement of the piston per minute in an engine, diameter of cylinder 18 ins. and length of stroke 2 ft.; revolutions per minute, 70.

$$\text{Then } (1{\cdot}5 \times 1{\cdot}5 \times {\cdot}7854) \text{ sq. ft.} \times 2 \text{ ft.} \times (70 \times 2) \text{ strokes}$$
$$= 494{\cdot}76 \text{ cub. ft. per minute.}$$

Piston rods are subjected to alternate pushing and pulling stresses which occur in rapid succession, and which must severely test the material of the rod, and they are now invari-

ably made of steel. The weakest part of the rod is at the screwed end which takes the nut. This part, however, is only subject to tension, and not to alternate tension and compression, for when the steam enters the cylinder underneath the piston (fig. 43) the whole load is carried by the screwed part of the piston rod ; but on the return stroke, when the piston is descending, the stress is removed from the screwed part and comes on the tapered part of the rod and the collar.

The load to be carried by the piston rod equals the difference between the pressure on the two sides of the piston. Thus, in a condensing engine the effective pressure per sq. in. on the piston equals the boiler pressure by gauge, plus 15 lbs. pressure of atmosphere, minus loss of pressure between boiler and cylinder, minus back pressure due to imperfect vacuum in condenser.

Crossheads and Guide Blocks

The crosshead forms a head at the outer end of the piston rod, to which the connecting rod is attached by a pin passing through the crosshead. It varies very considerably in design. Guide blocks are sometimes attached to each end of the pin, on either side of the crosshead, as in fig. 48. Another arrangement is to make a foot solid with the crosshead, which acts as a guide block, and works between guides, as shown in fig. 49.

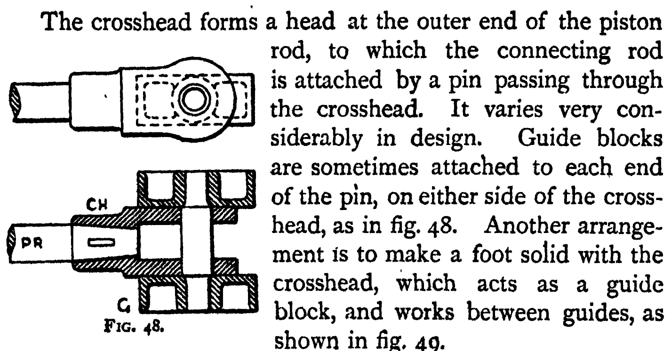


Fig. 48.

The blocks and guides prevent the oblique thrust or pull of the connecting rod from bending the piston rod. This can be seen by reference to fig. 50. When the piston P is being impelled forward, so that the rotation of the crank pin is in the direction of the arrow, the resistance at the crank pin *c* causes a downward thrust through the connecting rod *c g*, which may be resolved into two forces, one tending to compress the piston rod and the other to bend it in the direction T, causing a down-

ward thrust upon the guides. Again, when the piston is being driven back by the steam, the resistance of the crank pin at c' causes a downward pull at the point g of the piston rod, the tendency again being to cause a downward thrust upon the guides. If the engines were reversed the whole of the condi-

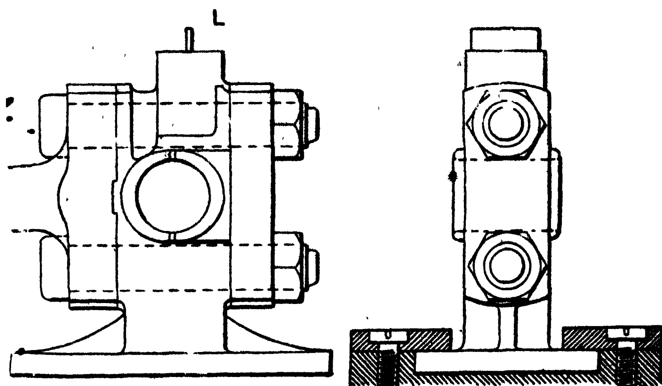


Fig. 49.

tions would be reversed, and the thrust gT would be upwards instead of downwards. Hence the prevailing direction in which horizontal engines should run is that shown by the arrow in the figure, so that the pressure on the guides should be upon

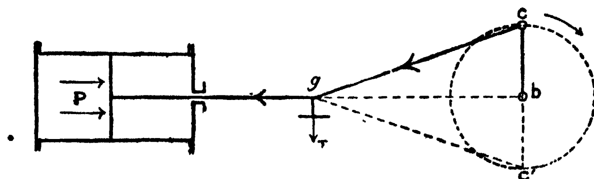


Fig. 50.

the lower rather than upon the upper guide bar; this is especially important for the sake of efficient lubrication.

It should be noticed that when the crank pin drags the piston, as it does, for example, when steam is shut off while the engine continues to rotate, the direction of the thrust on the guides is

reversed ; hence the necessity for a top and bottom guide bar under all circumstances. The amount of the thrust on the guides varies according to the angularity of the connecting rod, being greatest when the crank is at right angles to the axis of the piston rod, and being reduced to nothing at each end of the stroke; hence the guides wear hollow in the middle, and arrangements should exist for removing the guides and truing them up.

The amount of the thrust on the guides in the middle of the stroke may be found from the following simple formula :

$$\text{Maximum thrust} = \frac{\text{Pressure on piston} \times \text{radius of crank in ins.}}{\text{Length of connecting rod in ins.}}$$

Example.—Find the maximum thrust on the guides when pressure on piston at half-stroke = 20,000 lbs. ; radius of crank = 15 ins. ; length of connecting rod = 5 ft.

$$\text{Maximum thrust} = \frac{20{,}000 \times 15}{60} = 5{,}000 \text{ lbs.}$$

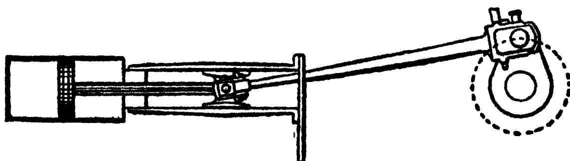


FIG. 51.
(Dotted circle=crank-pin path.)

When engines are required to rotate in either direction equally, as the locomotive, the surfaces in contact between the block and the guide are made equally large, as is the case in fig. 52, with the top and bottom guide bar; but when the engine is intended to rotate always in one direction, or nearly so, as in the marine engine and in factory engines, the surface on which the thrust comes is made sufficiently large, while the opposite surface may be much reduced, as is the case with the *slipper, or shoe guide* (fig. 49), the prevailing direction of the thrust being taken on the largest surface of the block.

THE CONNECTING ROD

The connecting rod connects the crosshead with the crank pin, and by its means the reciprocating or to-and-fro motion

of the piston is transformed into the rotatory or circular motion of the crank pin.

The length of the connecting rod, which is measured from the

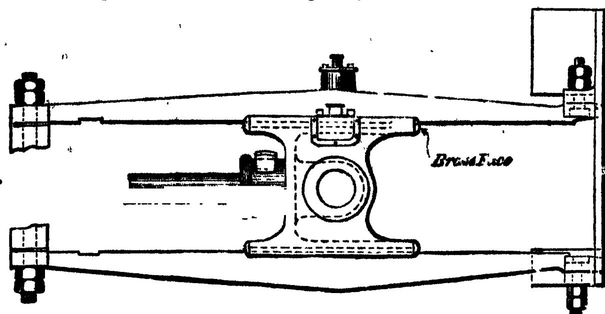


Fig. 52.

centre of the crank pin to the centre of the crosshead pin, varies from two to three times the length of stroke of the engine. By the ***stroke*** is meant the distance travelled by the piston from one

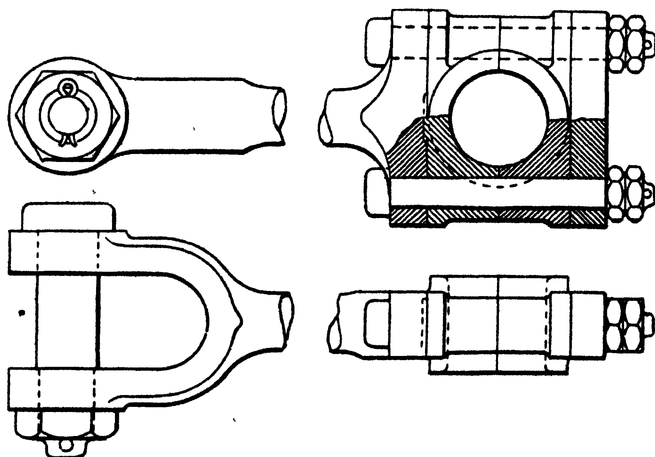


Fig. 53.

end of the cylinder to the other, which is equal to the diameter of the crank-pin path, or to twice the length of the crank arm.

Fig. 53 is an illustration of the marine type of connecting rod.

Fig. 54 shows a 'strap, gib, and cotter' arrangement for a connecting rod end.

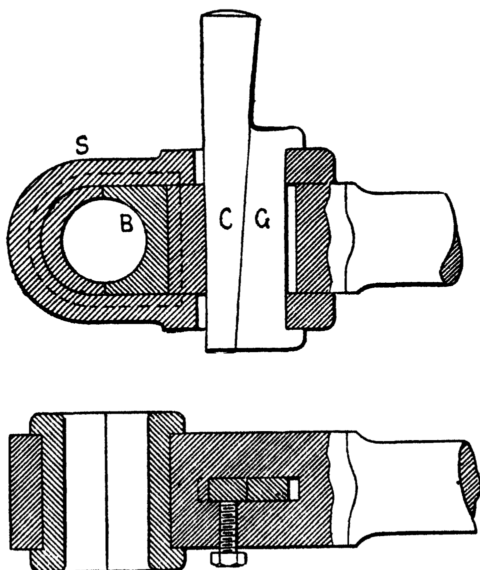


FIG. 54.
S, strap ; G, gib ; C, cotter.

Relative positions of piston and crank pin.—When the piston P is at either end of the stroke, the centre line of the

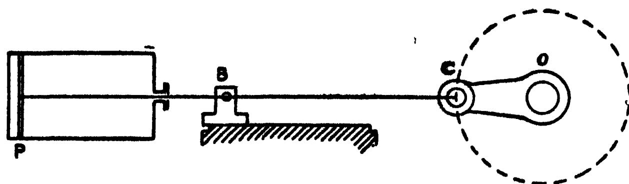


FIG. 55.

connecting rod B C and of the crank *o* C lie on the axis of the cylinder produced (see figs. 55 and 56), and the crank is then said to be on its *dead centre* ; for if the engine come to rest in this position, it will remain at rest, even when steam is admitted to

the cylinder, because the pressure of the piston is felt merely as a thrust on the crank shaft main bearing, and it has no tendency to cause the crank to rotate. In such a case it is necessary to 'bar' the engine round by the fly wheel till the crank has moved off the dead centre, before admitting steam against the piston. There are two 'dead centres' in a revolution.

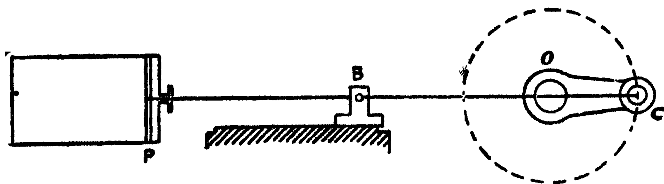


FIG. 56.

Let the dotted circle, fig. 57, represent the path of the crank pin about the centre of the crank shaft.

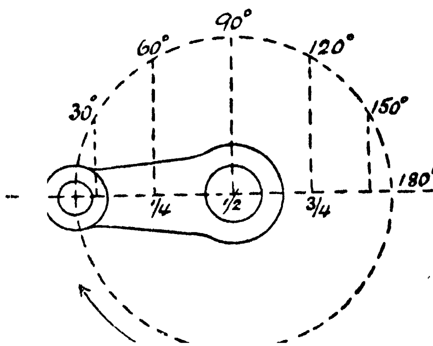


FIG. 57.

If the connecting rod were infinitely long, or if we neglect the obliquity of the connecting rod, then, when the crank pin is at any position between 0° and 180°, the corresponding position of the piston is found by dropping a perpendicular upon the diameter as shown, which diameter may be taken to represent the stroke of the piston. In such a case, when the crank pin is at 90° or one-fourth of a revolution from 0°, the piston would be in the middle of its stroke; but this is not the case in practice, because of the obliquity of the connecting rod, as will now be shown.

Since the position of the crosshead corresponds exactly with the position of the piston, we may for the present purpose

suppose the connecting rod to take hold of the piston direct, without the intervention of the crosshead and piston rod.

First, to obtain the position of the piston for a given position of the crank pin.

In fig. 58 let C_1 C_2 be the diameter of the crank-pin path, and let the length of the connecting rod be $1\frac{1}{2}$ times the stroke, namely, $1\frac{1}{2}$ times C_1 C_2. From C_1, with radius equal to length of connecting rod, mark P_1, the position of the piston when crank is at C_1; also from C_2 with the same radius mark P_2, the position of piston when crank is at C_2. From any intermediate position on the circular crank-pin path, and with radius equal to length of connecting rod, cut the line of stroke P_1 P_2, then the intersection will give the corresponding position of the piston. Thus, when

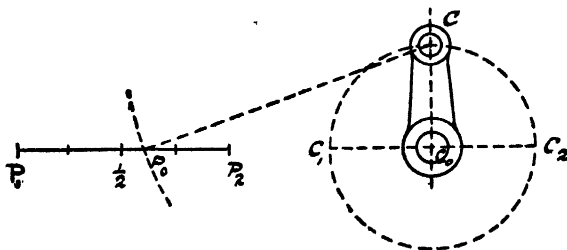


FIG. 58.

the crank pin is at C with the crank at right angles to the line of stroke, the piston position is not at half stroke, but at some position P_0 beyond half stroke; and the shorter the connecting rod the greater the distance travelled by the piston beyond the centre of the stroke; or, the longer the connecting rod the more nearly P_0 would coincide with the middle of the stroke.

From the figure it will be clearly seen that while the crank pin rotates at a uniform velocity through the first quarter of a revolution, the piston travels at the same time from rest at P_1 a distance P_1 P_0 greater than half the stroke, when its velocity is equal to that of the crank pin; and, during the uniform rotation of the crank pin through the second quarter, the piston travels a distance P_0 P_2, or less than half the stroke, again coming to rest at P_2.

Conversely, to obtain the position of the crank pin for a given position of the piston. When the crank pin is at C_1 (fig. 59), the piston is at P_1; and when the crank pin is at C_2 the piston is at P_2; and if the connecting rod were loose from the crank pin, and held at the centre of the crank shaft C_0 the piston would be at $P\frac{1}{2}$, namely, at the middle of the stroke. Now let the piston end of the rod remain in this middle position and move the other end of the rod from C_0 in an arc of a circle from centre $P\frac{1}{2}$ till it cuts the crank-pin path at C. Then C is the position of the crank pin when the piston is in the middle of the stroke. Any other position of the crank pin for a given position of the piston may be similarly obtained.

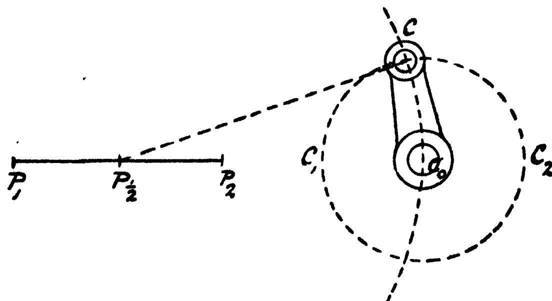


FIG. 59.

By the term *piston speed* is meant the *mean* speed of the piston. This, however, is less than the mean speed of the crank pin; for during one stroke of the piston the crank pin moves through a semicircular path, the length of which, compared with its diameter or the stroke of the piston, is as $1 : \frac{3{\cdot}1416}{2}$; or as $1 : 1{\cdot}5708$.

Thus, if the mean piston speed is 1,000 ft. per minute, the mean speed of the crank pin is $1000 \times 1{\cdot}5708 = 1570{\cdot}8$ ft. per minute.

By the principle of work, since the work done on the piston is the same as that done on the crank pin, and that the mean *speed* of the crank pin is 1·5708 times that of the piston, there-

fore the mean *pressure* on the crank pin in the direction of its motion is $\frac{1}{1 \cdot 5708}$ of the mean pressure on the piston.

Example.—In a direct acting engine the diameter of the cylinder is 17 ins., and the mean pressure of the steam is 60 lbs. per sq. in., the crank being 12 ins. long; what is the mean pressure on the crank pin in the direction of its motion? (Sc. and A., 1878.)

$$\text{Then mean pressure on piston} = 17 \times 17 \times \cdot 7854 \times 60 = 13614$$

$$\text{and mean pressure on crank pin} = 13614 \times \frac{1}{1 \cdot 5708} = 8670 \text{ lbs.}$$

Rotary engines.—Endless time and money have been spent on the invention of rotary engines, consisting for the most part of a piston which revolves within the cylinder, instead of one which moves from end to end of the cylinder, the object being to prevent the supposed loss of power due to the frequent changes of the direction of motion in the moving parts.

But, as a matter of fact, there is no such loss, for the energy exerted during the first half of the stroke in accelerating the piston from rest to its maximum velocity at the middle of the stroke, is restored to the crank pin during the latter half of the stroke in again bringing the piston to rest.

And, further, although the mean tangential pressure on the crank pin is less than the mean pressure on the piston, yet the distance through which the tangential pressure acts is greater than that through which the pressure on the piston acts in the proportion of 2 : 3·1416, namely, two strokes of the piston to one revolution of the crank pin, and by the principle of work (neglecting friction).

Pressure on piston $\times 2 =$ mean tangential pressure on crank pin 3·1416.

Hence there is no loss by the arrangement.

CHAPTER XI

THE SLIDE VALVE

BEFORE explaining the action of the valve, it will be helpful to the student to have a clear idea of the actual shape of the cylinder face. This is shown in the following diagram, fig. 60,

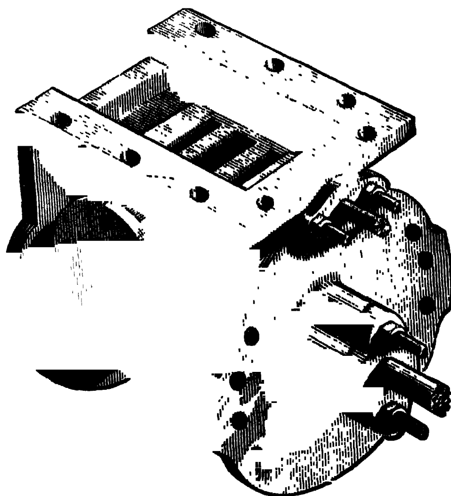


FIG. 60.

where the slide jacket cover and slide valve are removed so as to expose to view the long rectangular shaped ports in the cylinder face in the upper part of the diagram. Three rectangular openings are shown ; the middle port, which is the

exhaust port, is wider than the other two. It is a passage leading direct from the cylinder face to the outside of the cylinder, one end of the passage being the rectangular opening, called the exhaust port, and the other end the circular opening with a flange, shown in the figure, to which the exhaust pipe may be bolted. The two other ports are the steam ports—one leading to one end of the cylinder, and the other to the other end.

The slide valve is shaped somewhat like a hollow, rectangular, inverted dish; the edges of the dish, constituting the face of the valve, are planed and scraped to a perfectly true plane surface, and this works on a similarly prepared surface on the cylinder face.

The following diagram will explain the action of the slide valve. We will first take the simplest form of valve in which the edges of the valve are exactly the same width as that of the steam ports.

Fig. 61 shows such a valve in its central position completely closing both steam ports. The position of the piston at the same moment is at the end of the stroke, ready to commence a new stroke. The piston is connected to the crank pin C of the crank O C moving about the centre O of the crank shaft (shown out of its correct position for the sake of convenience), and the slide valve is connected to the pin E of the smaller crank O E moving about the same centre. The centre E is really the centre of the eccentric; but, as will be explained later on, the action is the same as though O E were a little crank. The dotted circles representing the paths of the crank pin C and of the centre of the eccentric E have their diameters equal to the stroke of the piston and valve respectively, and the positions C and E of these centres are correctly placed relatively to the positions of the piston and valve in fig. 61. The smallest movement of the shaft about its centre O in the direction of the arrow will cause the valve by its connection with E to uncover the left-hand port and admit steam against the piston.

Suppose the shaft to have described one-fourth of a revolution from the first position, then the new positions of the pins C and E, and of the piston and valve respectively, are

shown in the fig. 62. The distribution of the steam may be also followed by referring to the arrows, the steam being admitted from the boiler on one side of the piston, and on the other side exhausted into the air, or a condenser, by passing out through the hollow part of the valve into the exhaust passage.

As the piston continues to travel towards the end of its stroke, it will be seen, by following the movements of C and E, that the valve returns to its middle position, and again just closes the port as the piston reaches the end of the cylinder.

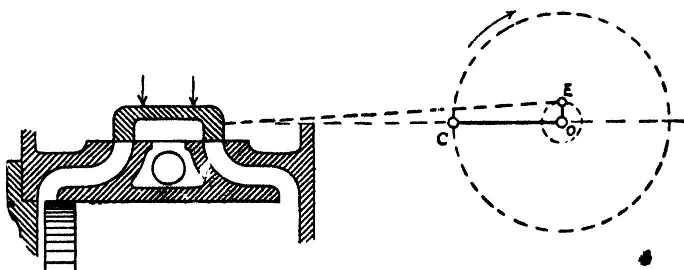


Fig. 61.

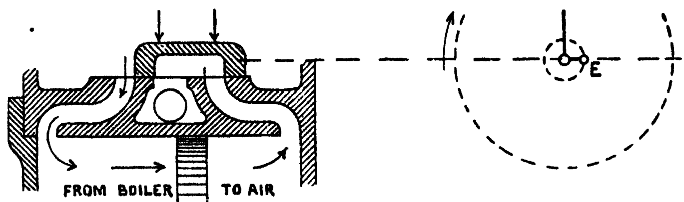


Fig. 62.

The valve then uncovers the right-hand port and the distribution of steam is reversed.

The valve which we have so far described has two important disadvantages :

(1) It admits steam to the cylinder throughout the whole length of the stroke of the piston. The waste of steam involved in not cutting off the supply at an early point of the stroke, and using it expansively, has been already pointed out.

(2) It opens the ports to steam and exhaust just *after* the

piston moves forward on its return stroke instead of just *before* it commences to return.

These disadvantages are overcome in two ways: (1) by adding *lap* to the valve – that is, by extending the width of its face—and (2) by giving it *lead*—that is, by causing it to move forward so as to open the port just before the piston reaches the end of its stroke.

Definitions of lap and lead.—The amount by which the valve overlaps the edges of the steam port *when at the middle of its stroke* is called the *lap* of the valve.

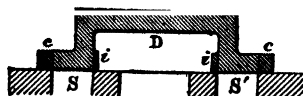


FIG. 63.

The amount by which it overlaps the outside edges is called the *outside lap.*

The amount by which it overlaps the inside edges is called the *inside lap.*

Thus, in fig. 63, the lightly shaded part shows the valve with no lap. The darker parts show the addition of outside lap *c i* and inside lap *i i*, by increasing the width of the face from the width of the port S to the width *c i.*

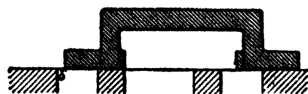


FIG. 64.

The amount of opening of the port for the admission of steam *when the piston is at the beginning of its stroke* is called the *lead* of the valve (pronounced *leed*). Thus the opening *b* (fig. 64) is the lead of the valve, if the piston at this moment is at the beginning of its stroke.

It will be noticed that, the inside lap *i* being less than the outside lap *c*, the lead to the exhaust port is greater than that to the steam port, which permits of a ready escape to exhaust.

When a valve has no lap, it moves on each side of its middle position, in order to open the steam port fully, a distance equal to the width of the port. In other words, the radius O E (fig. 61)=width of port. But, when lap is added to the valve (fig. 65), the distance moved on each side of its central position must be increased, if the port is to be fully opened, to the width of the port plus the lap. Hence the radius O E (fig. 65) repre-

senting the eccentricity of the eccentric or the half travel of the valve = width of port + lap.

Let the piston be situated at the beginning of the stroke (fig. 66); then, to admit steam to the cylinder, the valve must be moved forward from its middle position *a*, past the edge of the port *b*, until it has opened the port by a distance equal to the lead required, namely, *b c*. To accomplish this, the centre of the eccentric E must be moved forward to some position E′, making an angle C O E′ with the crank greater than a right

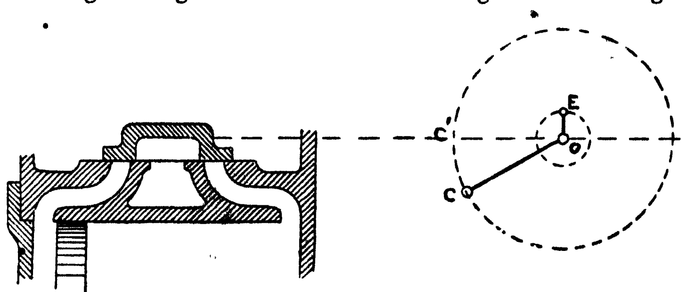


FIG. 65.

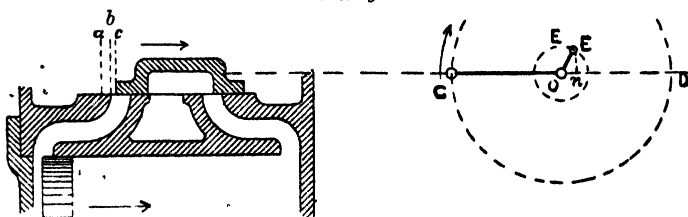


FIG. 66.

angle. To find this position: From the centre O on the centre line C D set off O *n* equal to *a c*—that is, equal to the lap plus the lead—and from *n* raise a perpendicular *n* E′ to cut the circular path of the eccentric centre. Then E is the position required, and O E′ produced is the centre line of the eccentric (see also fig. 67).

But the piston is assumed at the end of its stroke, therefore O C is the position of the crank, and we now have the relative positions of the crank and eccentric centre lines.

The angle E O E′ which the centre line of the eccentric is moved through beyond 90° ahead of the crank is called the *angular advance* of the eccentric. (See also fig. 67.)

Example.—The width of a steam port is 1 in., the lap of the valve $\frac{1}{2}$ in., and lead $\frac{1}{8}$ in. Find the eccentricity of the eccentric, and the angular advance of the eccentric.

Eccentricity of eccentric = half travel of valve
Half travel of valve = lap + port opening.
$= \frac{1}{2} + 1$ in.
$= 1\frac{1}{2}$ in.

Therefore, from centre *o*, with radius O E = $1\frac{1}{2}$ in., draw a circle representing the path of the centre of the eccentric. (Fig. 67, half size.) Let O C be the position of the crank, and draw *o* E at right angles to C *o*. On C *o* produced make *o a* = $\frac{1}{2}$ in. and *a b* = $\frac{1}{8}$ in., and from *b* draw *b* E′ perpendicular to C *o* to cut the path of the eccentric centre in E′. Join O E′. E O E′ is the angular advance of the eccentric.

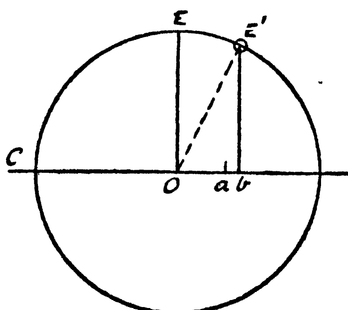


Fig. 67.

The action of the valve on the face of the ports may be easily followed by drawing the ports, and marking off the valve on the edge of a piece of paper, and moving the valve on the ports as required.

The effect of *outside lap* is :

(1) To increase the travel of the valve ;

(2) To cut off the steam at some point before the end of the stroke ;

(3) To cause the eccentric to be moved forward on the shaft, which results in an early opening of the exhaust port.

The effect of *inside lap* is :

(1) To close the exhaust port at an earlier point in the stroke, producing compression of the steam at the back of the piston ;

(2) To delay the opening to exhaust.

To set a slide valve.—Put the crank alternately on its two dead centres. Measure the opening of the port to steam allowed by the valve at each end of the stroke. When these

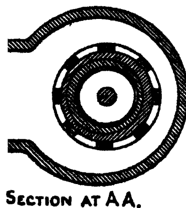
are equal to the lead allowed in each case, the valve is correctly set.

The travel of a slide valve from end to end of its stroke is equal to twice the distance moved by it on each side of its middle position $= 2 \times$ (outside lap + maximum opening of steam port).

Example.—Find the travel of a valve having $\frac{1}{2}$ in. outside lap, maximum opening of steam port $1\frac{1}{8}$ in.

$$2 \times (\tfrac{1}{2} + 1\tfrac{1}{8}) = 3\tfrac{1}{4} \text{ ins.}$$

Piston valves.—Fig. 68 illustrates the type of slide valve known as the piston valve, so called because it consists of two pistons, each working in a short barrel, in which an opening extending right round the barrel acts as the steam port. The chief advantage of the piston valve is that it is in equilibrium, there being no pressure of the valve against the cylinder face, as with the common flat or locomotive type of valve. It is therefore much used for the high-pressure cylinder of triple expansion engines. The pistons are each fitted with spring rings. In the example given in fig. 68, which is taken from a drawing kindly supplied to the author by Messrs. Bow, MacLachlan & Co., of Paisley, the piston rings are of phosphor bronze. There are two rings in each groove made eccentric, and one inside the other, as shown in the section A A. The rings are prevented from catching in the ports by diagonal bars across the ports, as


SECTION AT AA.

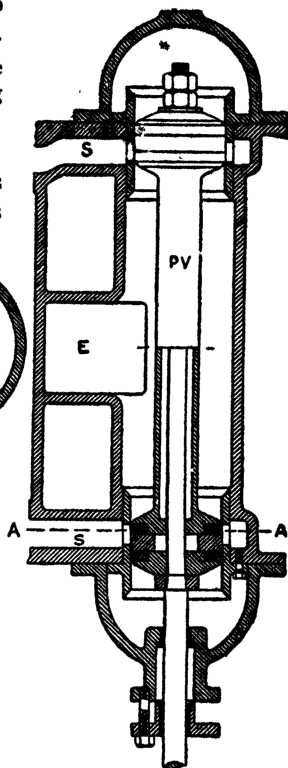


FIG. 68.

shown also in the section. The face of each piston is the same as the length of the face of the common valve, the inside and outside lap being also the same. The steam is admitted at the two ends of the valve, and exhausts into the space between the two pistons, and thence to the next cylinder.

Double-ported slide valve.—For large cylinders the travel of the valve, in order to open the port to supply sufficient steam, would necessarily be large. To reduce the travel and thereby also to reduce the work to be done by the eccentric in moving the valve, the double-ported slide valve is used as shown in fig. 69. The steam passage C of the cylinder terminates in *two* ports instead of one, and the steam ports are each made one-half the width of a single port, and therefore the travel of the double-ported valve is only half that of the common valve.

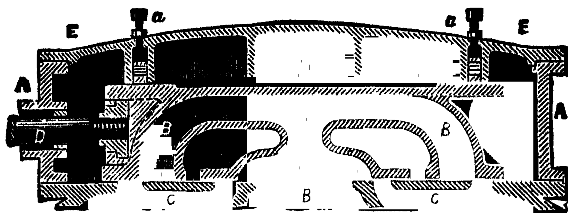


Fig. 69.

The valve is so constructed that, when in the middle of its stroke, each of the four steam ports is covered by it, the inside and outside lap in each case being the same as with the simple valve having the same travel, and hence its action in the distribution of the steam is exactly the same as with the simple valve. The arrangement is equivalent to two separate slide valves, the steam being supplied to the inner portion of the valve by steam passages in the sides of the valve. B B are the exhaust passages.

In large engines with a single flat slide valve, the pressure of the steam on the back of the valve would be so excessive, unless reduced by some means, that the load thrown on the eccentrics and working parts of the valve gear, owing to the friction between the valve and cylinder faces, would be enormous. To prevent this a packing ring is often fitted to

the back of the valve, as shown in the drawing (fig. 69). The ring is fitted in a circular groove in the slide jacket cover E, and it is tightened against a planed surface on the back of the valves by set screws *a a* pressing against a spring or packing. The set screws permit of a careful adjustment of the ring, so as to work steam tight without being excessively tight. By this arrangement the steam pressure is removed from a large portion of the back of the valve, and the enclosed space is connected instead by a pipe with the condenser.

ECCENTRICS

Eccentrics are used when a very small to-and-fro motion is required to be derived from a revolving shaft. They are applied mostly to drive steam-engine slide valves, or pumps having a short stroke. The simplest form of eccentric is a circular solid disc called a sheave, secured to and revolving with the shaft, the centre of the disc being 'out of centre' or 'eccentric' with the centre of the shaft. This arrangement is equivalent to a small crank (fig 70), the length of whose arm *r* is the same as the distance *c e* between the centre of sheave and centre of shaft. This length *c e* is called the *eccentricity* of the eccentric. The travel of the valve is equal to twice the eccentricity of the eccentric. The sheave is surrounded by a thin metal hoop, or band S (fig. 71), called the *strap*, to which the eccentric rod is attached. The rotation of the sheave H about the centre of the shaft is transmitted through the strap and rod, and results in the to-and-fro motion of the valve. The sheave may be considered as a very large crank pin, and the eccentric rod and strap as an ordinary

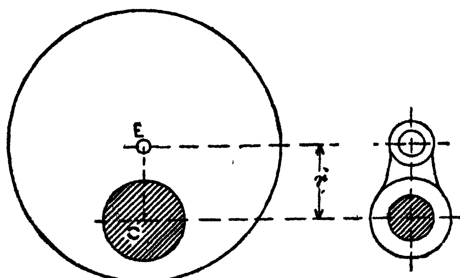


FIG. 70.

connecting rod. The sheave rotates within the strap just as the crank pin rotates within the head of the connecting rod.

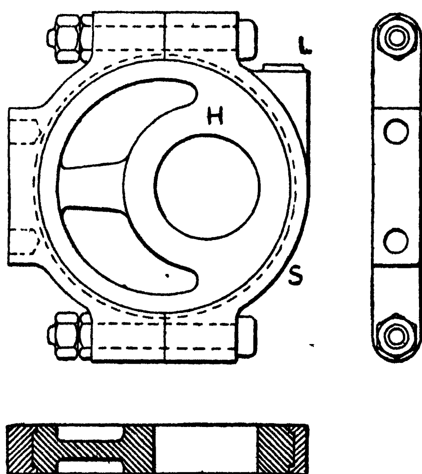


Fig. 71.

In order to get the eccentric in its place on the shaft it is mostly necessary to make the sheave in halves. The halves are secured together by two bolts, not shown, which are passed through holes drilled in the sheave and secured by split cotters. The strap is also made in halves, each half having lugs to take the bolts which secure them together. A small oil cup L is cast solid with the strap.

The sheave is secured to the shaft by a key fitting in a keyway cut in the shaft and in the sheave.

Reversing Gear—The Link Motion

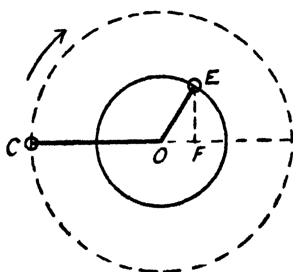


Fig. 72

Not the least important quality possessed by the steam engine is the ease with which it lends itself to the most perfect control. For by the movement of a handle the massive engines of a steamship running at a high speed may be instantly stopped and as quickly reversed. The following simple diagram will explain the principle of reversing gears.

It has been shown that when the crank is in some position

OC (fig. 72), the centre line of the eccentric will be in a direction OE ahead of the crank, the direction of rotation being shown by the arrow.

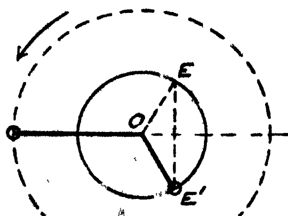


Fig. 73.

But suppose we wished to reverse the engine—in other words, to change the direction of rotation, as in fig. 73—then, unless we have some means of shifting the eccentric from E to E′, the engine will not reverse, but will only rotate one way.

This difficulty is easily overcome by the *link motion*, which is one of the most common methods of reversing, and it is done in the following way: Two eccentrics are used, one having its centre at E, and the other at E′, and by means of the link we have the power to use which eccentric we please, and to throw the other out of gear; hence the engine can be made to rotate in either direction with the greatest ease. Each eccentric is attached by a rod to one end of the slotted bar or link shown in fig. 74, and the link is moved transversely by the levers so as to bring the slide valve under the influence

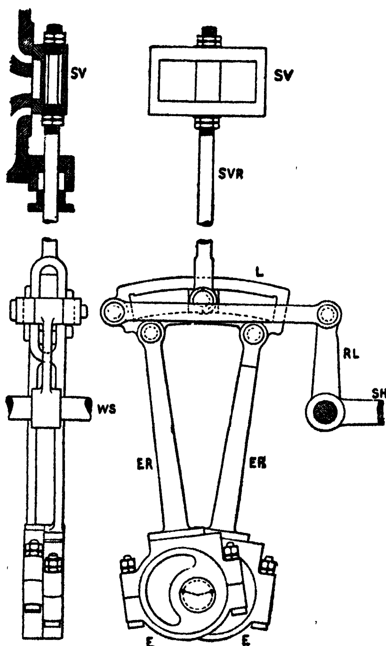


Fig. 74.
S V, slide valve; S V R, slide valve rod; L, link R L, reversing lever; S H, starting handle; W S, weigh shaft; E, eccentric; E R, eccentric rod.

of either eccentric as required. The slide valve is attached to a little block which fits in the slot of the link, so that any movement of the link in the direction of the axis of valve rod affects the position of the valve.

When the block is in the middle of the link, the valve is influenced equally by both eccentrics, with the result that the engine will not run in either direction. The nearer the block is to its mid position in the slot, the less is the travel of the valve and the earlier the steam is cut off in the cylinder. The link motion is therefore useful, not only for reversing but as an arrangement for working the steam expansively in the cylinder by varying the point of cut-off.

CHAPTER XII

*CRANKS AND CRANK SHAFTS.**

CRANKS are used to convert the reciprocating motion of the piston into circular motion. Fig. 75 shows two views of a simple overhanging crank.

This crank consists of an arm with a boss at each end—one to take the main shaft, and the other the crank pin. The crank is secured firmly in its place on the shaft, either by keying alone, or by 'shrinking' and keying. The shrinking is done by boring out the hole a shade smaller than the shaft, then heating the crank round the hole, and thus causing the material to expand and the hole to become larger. The crank is then slipped in its place on the shaft, and on cooling it contracts and grips the shaft tightly. Forcing on by hydraulic pressure is now frequently adopted in preference to shrinking on. The crank pin is shrunk in position or forced in by hydraulic pressure, and riveted over the end as shown.

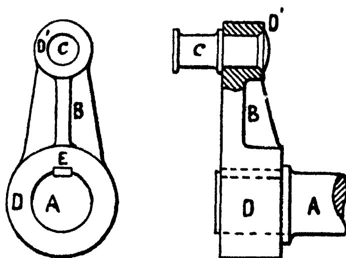


FIG. 75.
A=crank shaft; C=crank pin; B=web; D, D'=bosses; E=key.

The radius of the crank arm is measured from the centre of the shaft to the centre of the crank pin. The *throw* of the crank is equal to the diameter of the crank-pin path, and to the stroke of the piston.

The following is a crank axle for a locomotive with inside

cylinders, showing the cranks at right angles. The webs are here shown strengthened by wrought-iron straps shrunk on.

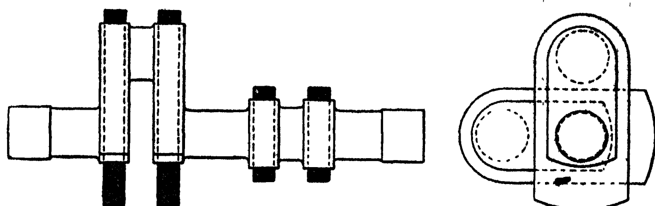


FIG. 76.

With such a shaft the engines will start in any position; for, if one crank is on its 'dead centre,' the other is in the best possible position for starting.

Examples of crank shafts are also given in figs. 36, 105, &c.

TANGENTIAL PRESSURE ON THE CRANK PIN

The tangential pressure on the crank pin is that share of the total pressure on the pin which tends to turn it about the centre of the shaft.

To present this subject in the simplest form we will suppose the pressure of the steam on the piston uniform throughout the stroke, the connecting rod to be of infinite length, or, in other words, to act always parallel to the centre line of the engine, and the moving parts to be without weight.

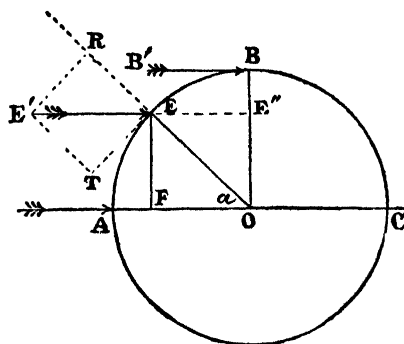


FIG. 77.

In fig. 77, A B C represents the path of the crank pin. Let P =the uniform pressure on the piston, and let O A the radius of the circle be chosen equal to P to any scale. When the

crank is in the position O A, the pressure P acts towards the centre only, and there is no tendency to *turn* the crank about the centre, but only to press the shaft against the bearing; hence in this position the tangential pressure is nothing. The same is true of the position O C, and O A and O C are termed the 'dead centres.' At the position O B of the crank at right angles to the direction of the force, the whole of the force is expended in turning the pin about O, and the tangential pressure on the pin is therefore equal to P. Between these two points A and B the tangential pressure varies from nothing at A to a maximum at B, and again falls to nothing at C. To find the tangential pressure at an intermediate point, as E, the force P may be resolved into two forces, one acting towards the centre of the shaft, and one perpendicular to it, or tangential to the crank-pin path at E. Thus at E draw the line E E′ parallel to A C and equal to the force P to scale, and by the parallelogram of forces resolve it into two forces, T E tangentially and R E radially. Then the line T E measures to scale the share of the force P acting at E, tending to turn the pin about the centre of the shaft, and R E measures also the share of force pressing the crank on the bearing. But when O E = E E′, the perpendicular E F is equal to T E for any position of E. Therefore for any point on the crank-pin path, under the conditions above described, a perpendicular let fall from it upon the diameter A C represents the tangential pressure on the pin at that point.

The diagram (fig. 78) further illustrates the variable nature of the turning effort on the crank pin, the amount of the turning effort at points 1, 2, &c. being proportional to the perpendiculars 1 a^1, 2 a^2, &c., and varying from nothing at A, 1 a^1 at 1 and so on, to a maximum at B, and again gradually falling to nothing at C.

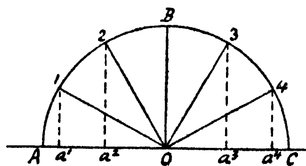


Fig. 78.

Further, it will be seen that this variation of twisting stress in the crank shaft occurs twice in every revolution of the crank

also that by increasing the pressure on the crank pin, either by increasing the area of the piston or the pressure of steam, the variation in the twisting stress is also increased in the same proportion.

If a pair of engines of equal power work on to one crank shaft, and the cranks are placed at angles of 0° or 180° with each other—that is, with the cranks together or exactly opposite—the twisting stresses on the shaft will be double those produced by the single engine alone, also the maximum and minimum twisting stresses on each crank will occur at the same time. But if the cranks be placed at right angles with each other, then, with the same engines, the maximum stress due to one engine will occur at the same time as the minimum stress due to the other engine, so that the total maximum stress will be reduced, and there will also be a much more uniform distribution of the stresses in the shaft. This will be more clearly seen by referring to the following figures.

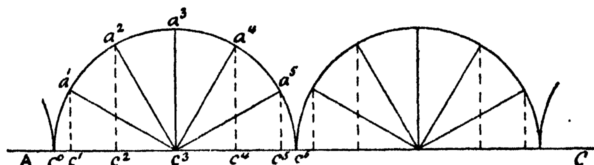


FIG. 79.

Fig. 79 is a continuous diagram of the turning effort on the crank pin for a single engine, the value of which at any point a^1, a^2, &c., is given by the vertical ordinate $a^1 c^1$, $a^2 c^2$, and so on, varying from nothing at c^0 to a maximum $a^3 c^3$ at a^3.

Fig. 80 shows the effect of the addition of another engine of equal power to the same shaft, when the cranks are at 0° or 180° apart. In this case the stresses are doubled throughout, varying from nothing to a maximum $a^3 b^3$, which is twice $a^3 c^3$ in fig. 79.

Fig. 81 shows the effect of placing the cranks at right angles to one another, the maximum turning effort as $b^0 c^0$ for one engine occurring when the turning effort of the other engine is nothing. The maximum stress is therefore never so

great as twice that due to the single engine, and the minimum stress never falls below the maximum due to one engine alone. There is, therefore, a much more regular distribution of the stress.

The uniformity or otherwise of the turning effort can be more clearly seen by setting up the ordinates from a horizontal

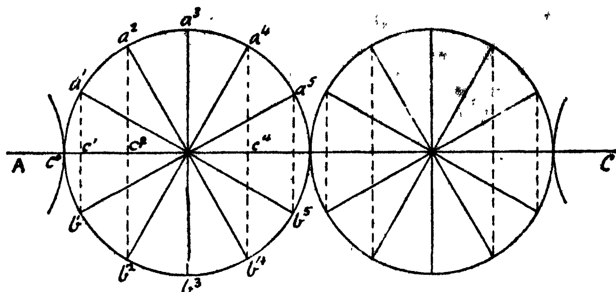


FIG. 80.

base, as in fig. 82. Thus, draw a horizontal line A C, and mark from A divisions A c, &c., equal and corresponding to the divisions on the semicircumferences, fig. 81. Then from these divisions (fig. 82) set up $a' c = a' c'$ (fig. 81), and $c b'$ (fig. 82)

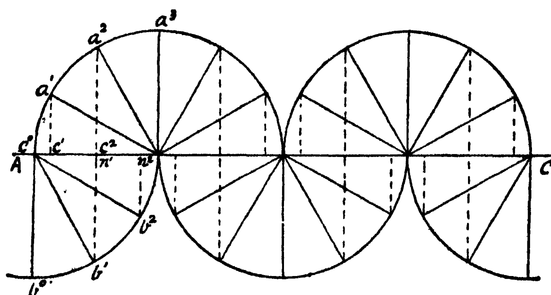


FIG. 81.

$= b' n'$ (fig. 81), and so on, and join the free ends of the lines. Then the total breadth of the figure gives the combined turning effort on the shaft. The variation in the stress may be still more conveniently represented by constructing the whole figure above the line A C as shown by the dotted parts ; thus

cb' is set off above $a' = a'd$, and so on; and the tops of the ordinates are joined by a free curve. The nearer this curve becomes to a horizontal line, the less the variation in the twisting stresses. For the treatment of this subject, when account is taken of the varying pressures of the steam throughout the

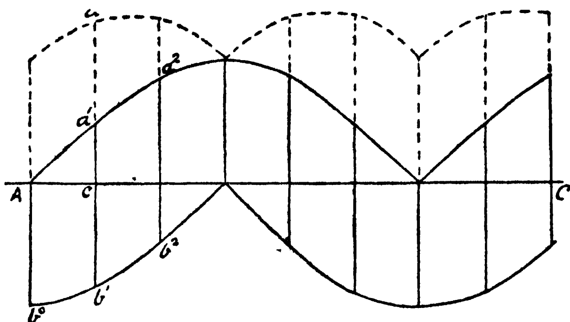


FIG. 82.

stroke, the obliquity of the connecting rod, and the weight of the moving parts, the student is referred to the work on the same subject in the Advanced Series.

Crank shafts having three cranks usually placed at 120° still further distribute the stresses, and cause a still more regular and uniform motion of the shaft.

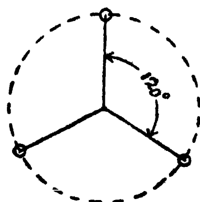


FIG. 83.

The variable character of the tangential or turning pressure on the crank pin is due to three causes:

(1) The communication of the pressure to the crank pin from a reciprocating piston through the connecting rod, the effect of which is that the tangential pressure varies from zero at the 'dead centres' to a maximum in the middle of the stroke.

(2) The expansive working of the steam by which the pressure falls from the beginning to the end of the stroke.

(3) The influence of the weight and velocity of the reciprocating piston, piston rod, and crosshead, which start from rest,

and are accelerated till they acquire a maximum velocity at the middle of the stroke, in accomplishing which a large portion of the steam pressure is absorbed, and is therefore not transmitted to the crank pin; while, during the latter part of the stroke, they are again brought to rest, the effect of which is to cause a

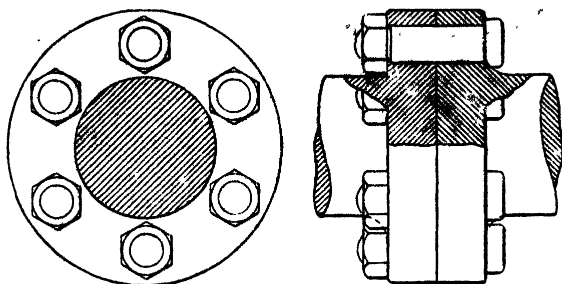


FIG. 84.

greatly increased pressure on the crank pin in addition to that due to the steam pressure on the piston.

It will be seen that the influence of the weight and of the velocity of the reciprocating parts tend to modify the variable nature of the stresses due to expansive working, and this is especially so at high speeds.

Shaft couplings.—The above diagram, fig. 84, illustrates

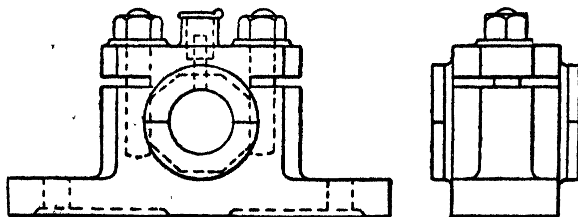


FIG. 85.

the method of joining lengths of shafting together at the ends. The ends of the shafts have flanges forged on them which are turned with the shaft and butt together end to end. Holes are drilled through the flanges, and they are firmly secured by bolts as shown.

Journals.—The journal of a shaft is that part of it which fits in the bearing (figs. 85, 86). It is of the greatest importance that the bearing surfaces of working parts should be sufficiently large. The length of the journal and of the bearing is proportioned so that the pressure per square inch on the bearing shall not be so great as to squeeze the oil out of the bearing, and so prevent proper lubrication. The length of the journal and bearing are increased for high speeds, and the bearing nearest the work is made longer than one further away from it.

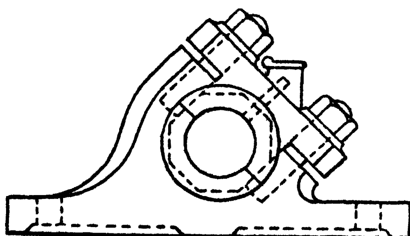


FIG. 86.

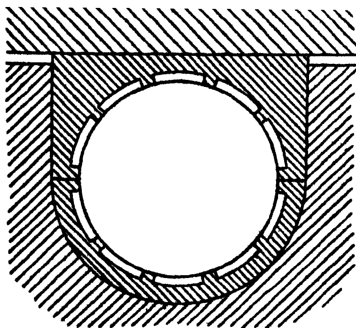


FIG. 87.

Pedestals, or Plummer Blocks (figs. 85, 86), consist of a body which holds the brasses, and a cap which is bolted down on the brasses to keep the bearing rigid.

When the resultant of the forces acting on the shaft is not vertical, but inclined at some angle with the vertical, the pedestal is constructed as shown in fig. 86.

For large engines the bearings are fitted with 'white metal,' as in fig. 87, on which shafts run more smoothly and with less friction and tendency to heat. The 'white metal' is run into grooves left for it in the brass.

CHAPTER XIII

CONDENSERS

THE condenser is a box or chamber into which the steam is passed and condensed after doing its work in the cylinder, instead of being exhausted into the air.

The object of the condenser is two-fold, being first to re-

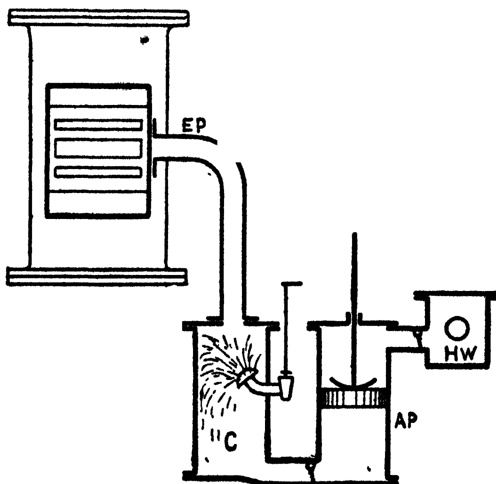
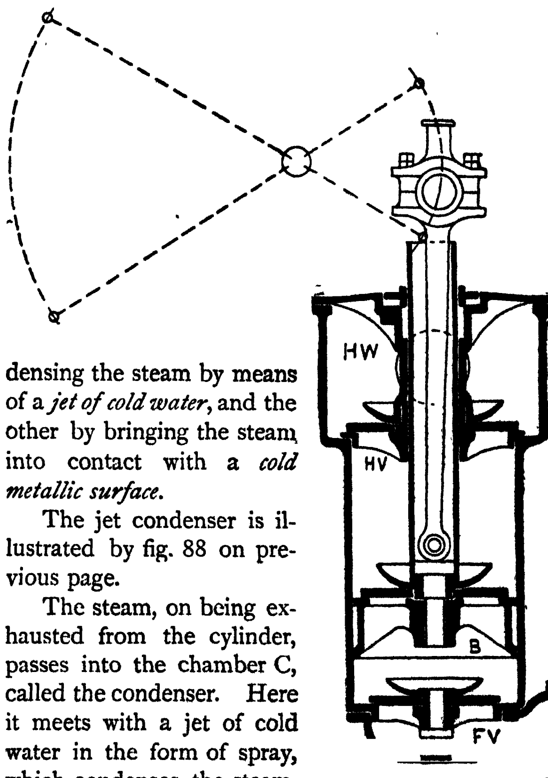


FIG. 88.
E P, exhaust pipe from cylinder; C, condenser; A P, air pump; H W, hot well.

move as far as possible the effect of atmospheric pressure from the back of the piston by receiving the exhaust steam and condensing it to water, thus creating a partial vacuum; and secondly to enable the steam which acts on the piston to be

expanded down to a lower pressure before leaving the cylinder than can profitably be done when the steam exhausts into the air.

There are two kinds of condensers, namely, the *jet* condenser and the *surface* condenser—the one, as its name implies, con-



densing the steam by means of a *jet of cold water*, and the other by bringing the steam into contact with a *cold metallic surface.*

The jet condenser is illustrated by fig. 88 on previous page.

The steam, on being exhausted from the cylinder, passes into the chamber C, called the condenser. Here it meets with a jet of cold water in the form of spray, which condenses the steam. The condensed steam and injection water must now be removed, and a pump A P is provided for the purpose. This pump is called the *air pump* because it removes, not only the water, but also the *air* which passes into the condenser mixed with the injection water, as well as the *vapour* which arises

Fig. 89.
H W hot well; B, air-pump bucket: H V, head valve; F V, foot valve.

from the water. It is the air and vapour in the condenser which are the cause of whatever *pressure* exists therein.

The condensed steam, injection water, air and vapour, are pumped into the *hot-well* H W, and thence to waste; and from the hot-well the water is taken to feed the boiler.

The suction valve of the air pump is called the *foot valve*, and the delivery valve is called the *head valve*.

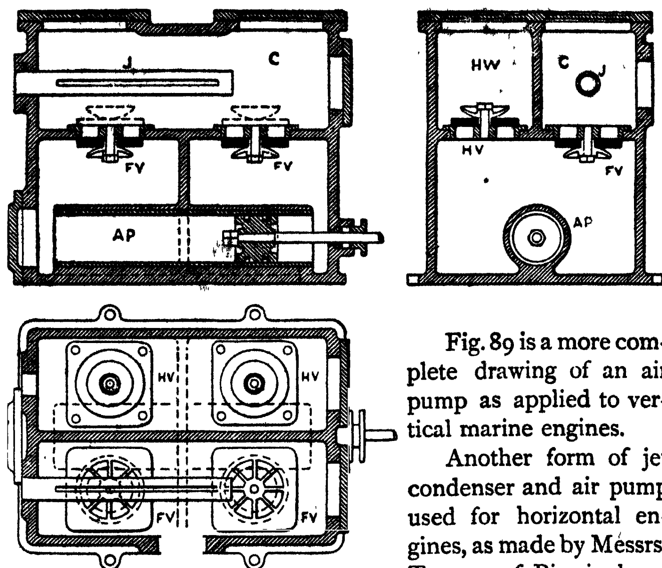


Fig. 90.

Fig. 89 is a more complete drawing of an air pump as applied to vertical marine engines.

Another form of jet condenser and air pump used for horizontal engines, as made by Messrs. Tangye, of Birmingham, is shown in the diagram, fig. 90, the air-pump rod being an extension of the piston rod.

The exhaust steam enters the condensing chamber C, where it meets with the cold-water spray J and is condensed. The condensed steam and injection water are removed from this chamber by the air pump A P, which draws it through the suction valve F V, and forces it forward through the delivery valve H V into the hot-well H W, from which the boiler feed may be taken. The remainder overflows.

The *surface condenser* has now entirely superseded the jet

condenser for marine engines as the natural consequence of the endeavour of marine engineers to increase the economy of their engines. In order to do this it was found necessary to increase the pressure of the steam used in the marine boiler, which up to 1860 was only about 30 lbs. on the square inch. Up to this time the boiler feed, which was from the hot-well of the jet condenser, was practically as salt as sea water, owing to the fact that the spray of the jet condenser was a sea-water injection, the sea water and the condensed steam being in the proportion of about 30 to 1. Even with the low boiler pressures the salt in the feed water was a serious drawback, for sea water contains $\frac{1}{33}$ of its weight of solid matter dissolved in it, and, when evaporated, the solids are deposited on the boiler plates, forming a more or less thick solid incrustation. This incrustation is a bad conductor of heat, and, further, since it keeps the water from contact with the hot furnace plate, there was great danger of the plate getting red hot and the top of the furnace collapsing. To prevent the water in the boiler from becoming too much saturated with salt, it was necessary to 'blow off' a portion of the water from time to time, and to supply its place with a fresh supply of ordinary sea water. By thus blowing away to waste large quantities of hot water, a considerable waste of heat was evidently the result.

But when the attempt was made to increase the pressure and temperature of the steam—now made possible by the introduction of steel plates for boiler construction—the difficulty arising from the presence of salt in the feed water became more serious, for with higher temperatures the solid matter is deposited much more readily, and its effects are far more mischievous. Hence the introduction of the surface condenser, which does away with the necessity of feeding the boiler with salt water; the condensed steam itself being pumped back again to the boiler as a fresh-water feed. For the steam is here condensed, not by being mixed with large volumes of cold water, but by coming into contact with cold metallic surfaces. The general arrangement of a surface condenser is shown in fig. 91.

The cold metallic surface required, by which to condense

the steam, is provided by means of a large number of thin tubes, through which a current of cold water is circulated. This arrangement supplies a large cooling surface within comparatively small limits of space.

The tubes are made to pass right through the condensing chamber, and so as to have no connection with its internal space. The steam is passed into the condenser and there comes in contact with the cold external surface of the tube, and is condensed, and removed, as before, by the air pump.

The condenser may be made of any convenient shape. It sometimes forms part of the casting supporting the cylinders of

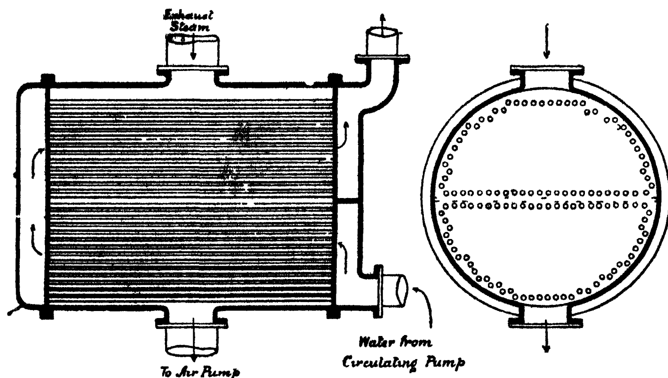


FIG. 91.

vertical engines; it is also frequently made cylindrical with flat ends, as in fig. 91. The ends form the tube plates to which the tubes are secured. The tubes are, of course, open at the ends, and a space is left between the tube plate and the outer covers, shown at each end of the condenser, to allow of the circulation of the water as shown by the arrows.

The cold water, which is forced through by a *circulating pump*, enters at the bottom, and is compelled to pass forward through the lower set of tubes by a horizontal dividing plate; it then returns through the upper rows of tubes and passes out at the overflow; the tubes are thus maintained at a low temperature. The steam enters at the top of the condenser and fills the

space surrounding the tubes. The tubes are made of brass, $\frac{3}{4}$ or $\frac{5}{8}$ inch outside diameter, and $\frac{1}{20}$ inch thick; and, being thin and of good conducting material, the steam is readily condensed against the cold outer surface of the tube.

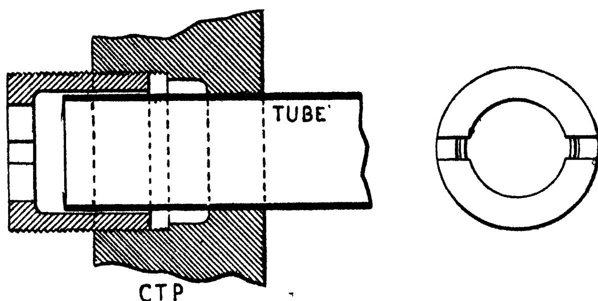


FIG. 92.
C T P = condenser tube plate.

The diagrams figs. 92 and 93 show two methods of connecting the tubes to the tube plates so as to make them tight.

Fig. 92 shows a little stuffing box and screwed gland, which is very generally used. The stuffing box is packed with tape or cord packing.

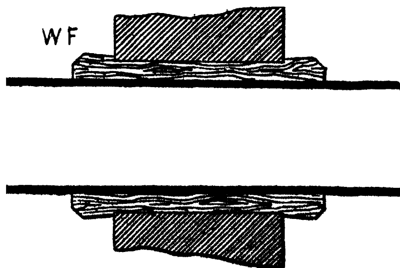


FIG. 93.

Fig. 93 is a wood ferrule W F made to fit the tube exactly, but a little too large for the hole. It is driven in between the tube and the hole in the tube plate. When in its place it absorbs moisture and swells, forming a perfectly tight connection.

Fig. 94 shows an enlarged view of a disc valve as used for air pumps. It consists of a grating covered by a circular disc of india-rubber, or, as in the figure, by a thin flat plate of phosphor-bronze (Coe & Kinghorn's patent). The water lifts the valve against the saucer-shaped guard, and passes through

the grating. When the water attempts to return, the valve closes down upon the grating and prevents it.

The *Vacuum Gauge* is used to determine the degree of vacuum in the condenser. It is graduated on the face from 0 to 30, and the degree of vacuum is indicated by a movable index-finger which passes over the graduated scale. The construction of the gauge is similar in principle to the Bourdon's pressure gauge described under Boiler Fittings.

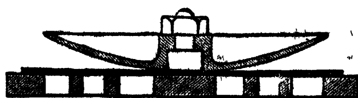


Fig. 94.

In order to be clear as to the meaning of the figures on the face of the vacuum gauge, it should be remembered that the object of the condenser is to remove as far as possible the pressure of the atmosphere from the back of the piston, and that the gauge is intended to show how far we are successful in doing this. The ordinary boiler pressure gauge indicates the pressure in the boiler above the atmosphere. If there were a partial vacuum in the boiler before the fires are lighted, the pressure gauge would not show it, and it only begins to indicate pressure when the pressure of the steam rises above the pressure of the atmosphere, this being the starting point or zero.

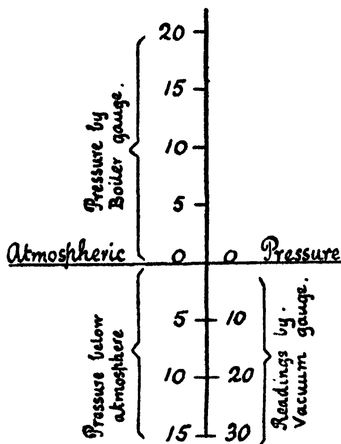


Fig. 95

The vacuum gauge also starts from the same zero, namely, the pressure of the atmosphere, and reckons backwards.

But, further, the figures on the vacuum gauge are doubled, and to understand the reason of this it should be remembered that they represent not pounds pressure but inches of mercury by the barometer, every *two* inches of mercury being equivalent

approximately to 1 lb. pressure, the old original vacuum gauge being constructed like a barometer. Hence, when the vacuum gauge indicates 25, it means that the difference between the pressure of the atmosphere and the pressure in the condenser is equivalent to the weight of a column of mercury 25 inches high, which is equal to $25 \div 2 = 12\frac{1}{2}$ lbs.; that is, 25 by the vacuum gauge means that the pressure in the condenser is $12\frac{1}{2}$ lbs. below the pressure of the atmosphere, or $15 - 12\frac{1}{2} = 2\frac{1}{2}$ lbs. absolute pressure opposing the piston, instead of 15 lbs. which would be the approximate back pressure due to the atmosphere if there were no condenser. The gain in horse-power by using the condenser may be calculated by the usual formula, $H\,P = \frac{PLAN}{33{,}000}$ where P is the gain of pressure by using a condenser, namely, in the present case, $12\frac{1}{2}$ lbs.

Pumps

The *feed pump* is used to feed the boiler, and it is required to supply a quantity of water at least equal to that evaporated and passed forward to the engine, together with leakage at safety valve, &c.; but to provide also for emergencies it is usually made capable of supplying from 2 to $2\frac{1}{2}$ times this quantity.

The feed pump is sometimes worked from the engine direct, or from the shaft by an eccentric attached to the plunger (see fig. 104). When it is worked independently of the main engine it is called a 'donkey pump.'

The following diagram, fig. 96, illustrates the construction of a simple feed pump. It consists essentially of a plunger P of a suction valve S and a delivery valve D.

The same construction may be used for the *bilge pump*, which pumps water that accumulates in the bilge or bottom of the ship.

The action of the pump may be explained as follows: Suppose the plunger P at the bottom of its stroke, and the whole interior of the pump to be full of air. When the plunger rises the pressure on the suction valve S will be reduced, and the air in the supply pipe will lift the valve and flow into the barrel. The pressure of the air in the supply pipe is

now less than before, and accordingly the pressure on the external surface of the water forces water up the pipe to such a height as to make the pressure inside the pipe balance the pressure outside. When the plunger returns the suction valve is closed by the pressure, and the air is forced out through the delivery valve D. Each time the stroke of the plunger is repeated, the water will rise in the supply pipe until at last it reaches and fills the barrel. Now, when the plunger returns, it forces water instead of air through the delivery valve.

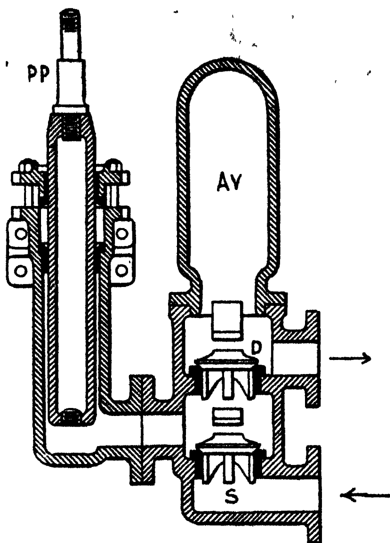


Fig. 96.

The height of the column of water which will balance the pressure of the atmosphere is 34 ft.; that is, a column whose weight is about 15 lbs. per sq. in. In practice, however, the supply can never be drawn from a depth greater than about 25 ft.

The valves are prevented from rising above a certain height by stops shown in the figure. The lift of a valve should not exceed one-fourth of its diameter, for with this lift the whole of the water which passes through the valve seating can escape freely round the edge of the valve. Any further lift is therefore unnecessary.

Thus, when the area of opening round edge of valve and the area of the valve are equal, we have

$$\text{area round edge} = \text{area of valve};$$
$$\text{dia.} \times 3{\cdot}1416 \times \text{lift} = \text{dia.}^2 \times {\cdot}7854;$$
$$\text{lift} = \frac{\text{dia.}}{4}.$$

Large valves are prevented by the stop from lifting so much as this because of the excessive knocking which would result.

Air vessels A V are chambers fitted to pumps close to and beyond the delivery valve, fig. 96. The air in the water collects in this vessel and forms a cushion or spring which enables the water to be delivered in a continuous stream instead of intermittently.

The capacity of a pump in cubic inches = area of end of plunger × length of stroke in inches.

The weight of a cubic foot of fresh water = 62·5 lbs., or 1,000 ounces.

The weight of a cubic foot of salt water = 64 lbs.

1 lb. of water occupies ·016 cub. ft.

1 gallon of water = 10 lbs.

CHAPTER XIV

GOVERNORS

A GOVERNOR is fitted to an engine for the purpose of securing, as far as possible, a uniform rate of rotation, and preventing variation of the speed at every fluctuation in the load or the boiler pressure.

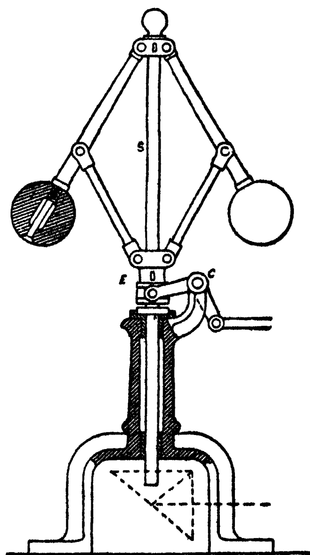


FIG. 97.

None of the governors applied to steam engines are able to accomplish this result *perfectly*; for, being themselves driven by the engine, they cannot begin to act until a change of velocity has first occurred.

In practice, however, the governor is an invaluable adjunct to the steam engine; for, when any change of velocity does take place, the governor instantly acts and prevents anything more than a small alteration of speed. Any permanent adjustment of the speed is regulated by hand at the steam supply.

The following is a description of the Watt Pendulum Governor. The study of this governor will serve to introduce the student to those principles of construction upon which this and most other governors are based.

The central spindle S of the governor, fig. 97, is made to rotate by means of a belt, or, better, by a small shaft driven from the engine shaft by bevel wheels communicating with the bevel wheels at the bottom of the spindle. The spindle, arms, and balls then all rotate together, and at the normal velocity of the engine the inclination of the arms is about 30° with the vertical. If the velocity of the engine increase, due to removal of load, the balls and arms open out from the spindle, and in doing so they lift the sleeve E, which slides up and down on the spindle. This movement is communicated by levers moving about the fixed fulcrum C, to the throttle valve, by which the passage

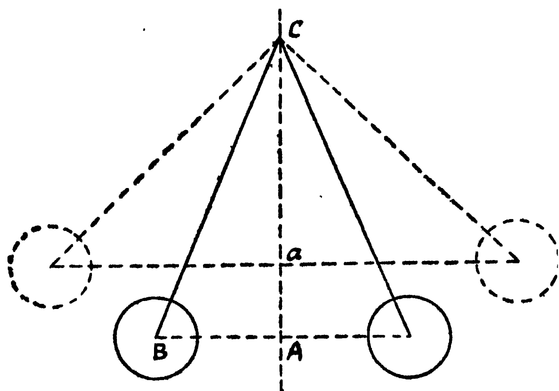


FIG. 98.

for the supply of steam to the engine is contracted ; or to an expansion gear, which is also an arrangement for reducing the steam supply, and the increasing speed of the engine is thereby checked. A slot is cut in the central spindle through which a cotter or pin secured to the sliding sleeve passes. The length of this slot limits the travel of the sleeve.

There are three forces acting on the governor balls during rotation, namely : the *weight* of the ball which acts vertically downwards, the *centrifugal force* which acts horizontally outwards, and the *tension in the arm* ; and these three forces are in equilibrium and are represented proportionally by the three

sides A C, A B, and B C (fig. 98), which are respectively parallel to the forces. The vertical distance C A is called the *height* of the governor or the height of the cone of revolution, and this height is constant for a given number of revolutions per minute.

The revolutions of the governor obey the same law as the oscillations of the pendulum, namely: the number of revolutions is inversely proportional to the square root of the height of the cone of revolution.

Thus, any change in the speed of the engine causes the governor balls to fly off from the centre, and a change in the

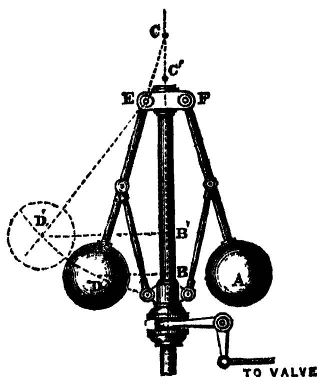


Fig. 99.

height of the governor to take place, as from C A to C *a*, fig. 98. It is the raising of the sleeve A to *a* by which the governor is made to influence the throttle valve or expansion gear; but, in order to open the throttle valve, it requires to be driven at an increased speed, and this is precisely what the governor is intended to check.

Such a governor, therefore, evidently permits of a variation in the number of its revolutions, and, therefore, also of the revolutions of the engine, between the limits due to the varying height C A of the cone of revolution. But a perfect governor would permit of no increase either in the number of

its own revolutions or that of the engine; and, although this ideal cannot be attained, still it is the aim of designers to reduce this variation in the height of the cone as much as possible; or, in other words, to enable the governor to lift a sufficient distance to close the valve without going through a considerable variation in speed in rising from its lowest to its highest position. The effect of the movement of the balls on the height of the cone when the point of suspension of the arms is on the centre line of the spindle is shown in fig. 98.

When, however, the arms are suspended from points E and F (fig. 99), not on the centre line of the spindle, and the balls rise from D to D′, the height of the cone now varies between C B and C′ B′, instead of between C A and C *a* as before, the effect being to still further increase the amount of variation in height, and, therefore, in revolutions of the engine, for a given lift of the sleeve. The points of suspension E and F should, therefore, be as near the centre of rotation of the spindle as possible.

The speed of the governor is independent of the weight of the balls, but the parts require to be sufficiently heavy to exercise proper control over the throttle valve or expansion gear.

Various forms of 'parabolic' governors have been introduced to give the necessary movement of the sleeve without the accompanying necessary increase of velocity.

The Watt governor is a slow-speed governor, owing to its height. To run at a higher speed it must be made much smaller, and then it would not be sufficiently powerful to control the supply of steam to the cylinder.

But the tendency of engine building has long been towards higher speeds, and for quick-running engines a Watt governor geared so as to run slower than the engine is not sufficiently sensitive. This governor is, therefore, now largely superseded by various forms of high-speed governors, of which the 'Porter' governor, illustrated by fig. 100, is one of the most common. This governor consists of two small balls with arms as before, but the lower links are jointed direct to the balls by means of a pin through the centre, their

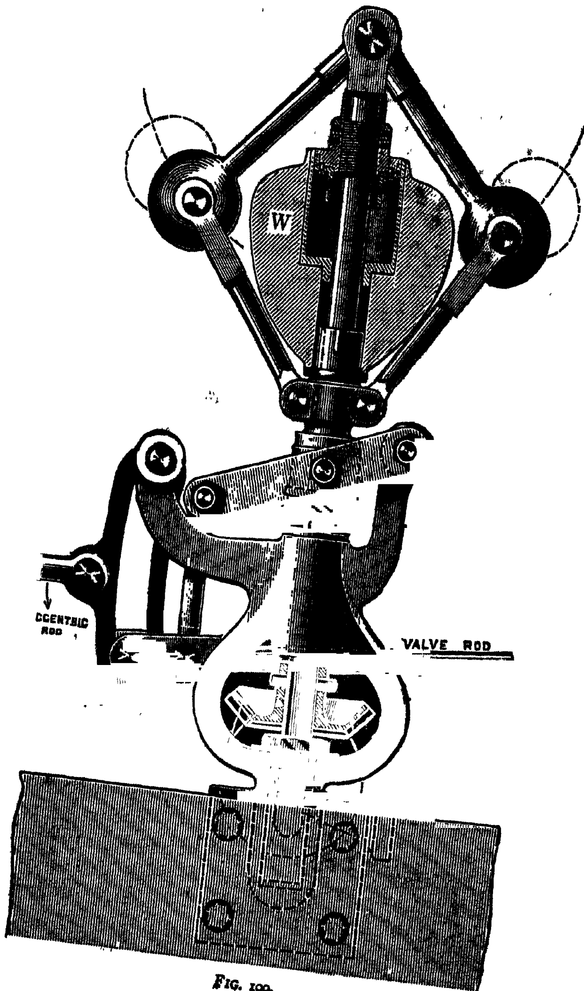


Fig. 100.

lower ends being connected with the sliding sleeve. Resting on the sleeve, and free to slide up and down the central spindle with it, is a weight W. This weight prevents any movement of the sleeve until the speed of the balls is such that their centrifugal force is sufficient to lift it. The governor has then the control of the engine. The heavier the central weight, and the smaller the balls, the higher the speed and the more sensitive the governor. The form of governor illustrated in fig. 100 is Tyrrel and Deed's Patent, made by Messrs Clayton and Shuttleworth of Lincoln. The special feature of this governor is the dash-pot put into the dead weight. The object of the dash-pot is to give steadiness to the governor.

The form of valve adopted when the governor is used for throttling the steam—that is, contracting the opening for supply—is the *double beat equilibrium disc valve*, illustrated in fig. 141.

In fig. 100 the governor is shown having an arrangement for regulating the travel of a cut-off valve on the back of the slide valve, instead of being connected with a throttle valve. The eccentric rod causes the link shown in the figure to oscillate about the upper fixed centre. The valve rod is attached to a sliding block in the link. When the speed increases sufficiently to cause the rotating balls to lift the weight and sliding sleeve, the end of the valve rod is raised in the link, and the travel is reduced, thereby cutting off the steam at an earlier point in the stroke.

FLY-WHEELS

The importance of a uniform velocity of the engine has been already pointed out.

But the turning effort on the crank pin, as we have seen, varies very considerably during each revolution; there is, therefore, a constant tendency to fluctuation of speed. In order to counteract this tendency the fly-wheel is added to stationary engines. The driving wheels answer the same purpose in locomotives.

When the turning effort on the crank pin during a portion of the revolution is greater than the resistance due to the load,

the speed of the engine is increased; and, conversely, when the resistance is greater than the turning effort, the speed of the engine is retarded.

The fly-wheel, owing to its great mass and to the distance of the mass from the centre of the shaft, resists very effectually all tendencies to changes of speed. For excess of turning effort, instead of causing an immediate and excessive change of velocity, is absorbed in giving a relatively small additional velocity to the mass of the rim of the wheel, and the power thus absorbed is restored when the turning effort falls below the resistance, thus maintaining a practically uniform velocity of the crank pin.

Locomotive Engine

The figures on pp. 126 and 127 illustrate the general construction and arrangement of an express passenger locomotive engine. The references to the parts are given below the figure.

It is necessary that the locomotive shall be self-contained—that is, it must consist of a boiler and an engine, and the whole machine must be placed upon one carriage. The problem for locomotive engineers is how to obtain the greatest possible power for the least possible weight. This is done by working at high steam pressures, using small boilers of great strength, and of high evaporative efficiency, and using the steam at high pressure in small cylinders in order to obtain a large amount of power with a comparatively light engine, economy in the use of steam being sacrificed in order to keep down the weight.

The engine and boiler are each bolted independently to the frame of the carriage. The frame is self-contained, and through it the whole of the stresses due to the pressure on the pistons, and the pull on the draw-bar due to the load, are transmitted.

The frame is carried on wheels, one arrangement of which is shown in the figure.

It will be noticed that the axle of the trailing wheels is placed just behind the boiler, the axle of the driving wheels just in front of the fire box, leaving clearance for the cranks and connecting-rod heads, and the axles of the bogie (or small

The Locomotive.

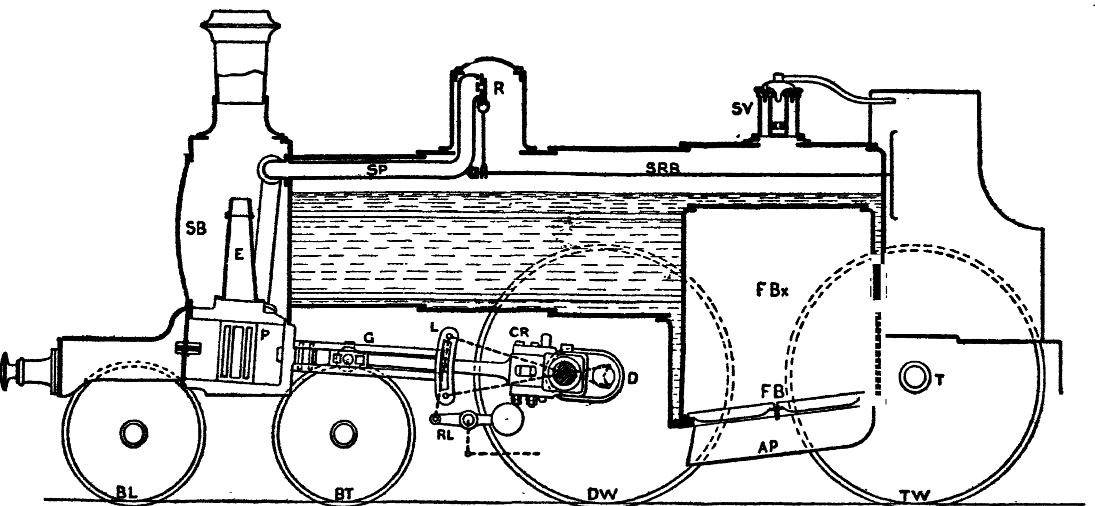


FIG. 101.

FBx, fire box; FB, fire bars; AP, ash pan; SB, smoke box; SV, safety valve; R, steam regulator valve; SRB, steam regulator valve bar; SP, steam pipe; E, exhaust pipe; P, steam ports; G, guides; L, link motion; RL, reversing lever; CR, connecting rod; C, cylinder; D, driving axle; T, trailing axle; DW, driving wheel; TW, trailing wheel; BL, bogie leading wheel; BT, bogie trailing wheel; CG, coupling rod.

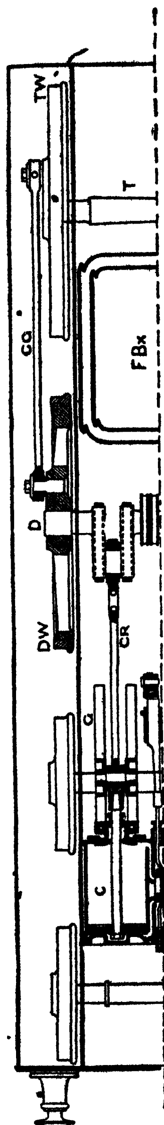


Fig. 102.

auxiliary carriage which works on a pivot beneath the cylinders) are placed in front of and behind the cylinders. The bogie wheels guide the engine, and prepare the rail to receive the weight of the large driving wheels ; the hind or trailing wheels steady the engine, while the driving wheels transmit the power of the engine to the rail, and they are placed as nearly as possible under the centre of gravity of the whole.

The locomotive boiler is described in detail under the heading of Boilers. The locomotive engine is similar in principle to that already described on p. 70, with the addition of the link motion for reversing.

The common arrangement is to have two cylinders of equal diameters, both using steam direct from the boiler, and exhausting independently into the chimney through the exhaust or 'blast' pipe, the cylinders having the several working parts of a complete engine, thereby forming a pair of engines acting on one crank shaft with the cranks at right angles. Compound locomotives are running on the lines of one or two English Railway Companies, and are said to give satisfactory results. The principle of the compound engine will be considered in the next chapter.

The cylinders of locomotives are constructed of the best, close-grained, hard and strong cold blast cast iron ; the pistons are made of good tough cast iron ; the piston-rods are best cast steel, tapered at the ends and secured to the piston by a gun-metal nut with a taper steel pin through the nut.

The valve spindles are of best Yorkshire iron, working through gun-metal bushes and glands in the steam-chest.

The crossheads are of the best Yorkshire iron, case-hardened; the sleeves are of the best hard cast iron. The gudgeon pins are of wrought iron, case-hardened.

The guide bars are of the best mild crucible cast steel.

The eccentric sheaves are in two parts, the smaller being of Yorkshire iron, and the larger of hard cast iron; the eccentric straps are of good tough cast iron; the eccentric rods are of Yorkshire iron, and the working parts and pins are case-hardened. The connecting and coupling rods are of Yorkshire iron; all cotters and bolts of mild steel.

The crank pins are of Yorkshire iron, case-hardened.

The following particulars of a compound locomotive goods engine were given in a paper read before the Institution of Mechanical Engineers by Mr. R. H. Lapage :—

	High pressure.	Low pressure.
Cylinder, diameter	16 ins.	23 ins.
Ratio of piston areas	1	2·1
Length of stroke	24 ins.	24 ins.
Length of connecting rod	6 ft.	6 ft.
Throw of eccentrics	$6\frac{1}{4}$ ins.	$6\frac{1}{4}$ ins.
Angle of advance, forward gear . . .	4°	4°
,, ,, back gear . . .	14°	14°
Travel of valve, full forward gear . .	$3\frac{1}{2}$ ins.	$3\frac{3}{4}$ ins.
,, full back gear . . .	$3\frac{13}{16}$ ins.	$3\frac{5}{8}$ ins.
Lap of valve	1 in.	1 in.
Steam ports	$1\frac{1}{4} \times 14$ ins.	$1\frac{5}{8} \times 17$ ins.
Cut-off, ordinary running	40 per cent.	50 per cent.

Pressure of steam in boiler, 175 lbs. per sq. in. above the atmosphere.

Exercise 1.—Find the area of the steam ports in each of the above cylinders, and express the ratio of steam port area and piston area in the two cases. *Ans.* H.P. cylinder 1 : 11·5 or 8·7 per cent.
L.P. cylinder 1 : 15 or 6·65 per cent.

Exercise 2.—The coal consumed in a compound locomotive was 79 cwts. in a run of 300 miles. The water used was 7546 gallons. Find the evaporation per lb. of coal.

Ans. $\frac{7546 \times 10}{79 \times 112} = 8{\cdot}5$ lbs. of water per lb. of coal.

CHAPTER XV

COMPOUND ENGINES

COMPOUND engines are those which have two or more cylinders of successively increasing diameters so arranged that the exhaust steam from the first and smallest cylinder is passed forward to do work in a second, and sometimes a third or fourth cylinder, before escaping to the condenser.

The compound engine enables the fullest advantage to be taken of the expansive power of high-pressure steam :

(1) By reducing the range of temperature in any one cylinder, and thereby reducing initial condensation of the steam in the cylinder.

(2) By taking advantage of the re-evaporation which accompanies cylinder condensation. For, since the bulk of the re-evaporation in a cylinder takes place during exhaust, it is obvious that in a single-cylinder engine the steam formed by re-evaporation during exhaust passes away to the air or the condenser to waste without serving any useful purpose. But when the steam is exhausted into a second or third cylinder, the steam formed by re-evaporation in one cylinder is utilised in doing useful work on the pistons of the succeeding cylinders.

(3) By the ease with which it may be adapted to work on to two or more cranks, thereby reducing the excessive variation of stress which occurs in a single-cylinder engine when working with steam at a high initial pressure expanded to a greatly reduced terminal pressure.

The following diagram (fig. 103) illustrates the difference between the action of the steam in a simple engine and in a triple expansion compound engine.

Suppose 1 lb. of steam at 150 lbs. pressure absolute admitted to a single cylinder and expanded down to 12 lbs. pressure absolute and exhausted into a condenser, when the pressure averages 3 lbs. absolute. Then the action of the steam in the single cylinder is represented by the whole figure shown cross-lined.

In such a case the temperature in the cylinder would vary from 358° F., the temperature of the steam at 150 lbs. pressure down to 142° F., the temperature of the steam at 3 lbs. pressure; or a difference of 358—142=216° F. between the initial

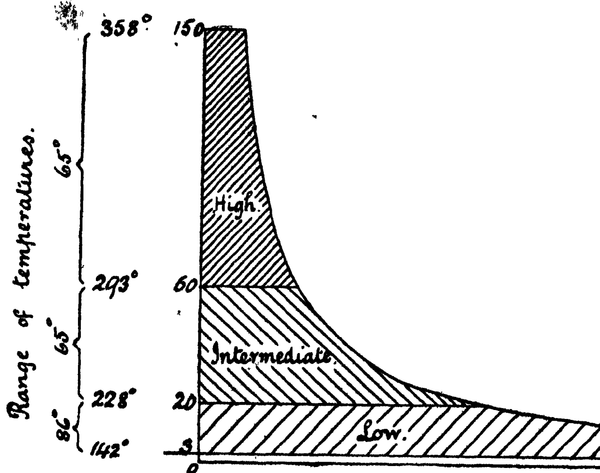


FIG. 103.

and final temperature in the cylinder. And since cylinder condensation increases with the increase in the range of temperature, the loss of steam by initial condensation would here be very great. If now the expansion of the steam be spread over three cylinders (called respectively high, intermediate, and low) the range of temperature in each will be proportionally reduced. Thus in the high-pressure cylinder, working between 150 lbs. and 60 lbs. pressure, there is a variation of 65° F.; in the intermediate cylinder, working between 60 lbs. and 20 lbs. pressure, there is again a variation of 65° F.; in the low-pressure

cylinder, working between 20 lbs. and 3 lbs. pressure, there is a variation of 86° F.

Again, the initial stress on the piston of the single-cylinder engine would be equal to forward pressure minus back pressure $=(150-3)\times$ area of piston, while the terminal stress would be $(12-3)\times$ area of piston; and therefore the initial stress is $\frac{147}{9}=16\cdot3$ times the terminal stress. This would be a most objectionable variation of stress on the working parts, and as the engine must be made strong enough to bear the maximum stresses due to the high initial pressure acting on a large piston area, a much heavier engine would be required than if the stresses were more judiciously distributed. If now the expansion of the steam, the range of temperature, the initial stresses, and the total work are distributed among three cylinders connected with three cranks, a much more economical and mechanically perfect engine is the result.

The shaded parts marked *high*, *intermediate*, *low*, represent the distribution of the work among three separate cylinders.

The diagram further illustrates the historical growth of the steam engine, for the bottom part of the figure represents the condition of the early engines working up to 20 lbs. pressure with a single cylinder; then came higher pressures, higher rates of expansion, and two-cylinder compound engines, and later, with the introduction of steel for boilers, and surface condensation, we have had a rapidly increased boiler pressure and rate of expansion, and the introduction of the three-cylinder or triple expansion compound engine. Pressures are still increasing, while the terminal pressure remains constant, and a fourth cylinder is in many instances now being added, forming a quadruple expansion engine.

Figs. 104, 105, and 106 illustrate a two-cylinder compound mill engine, H P being the high-pressure and L P the low-pressure cylinder. The steam passes from the boiler by the steam pipe S P into the valve chest of the high-pressure cylinder, where it is admitted to the cylinder and cut off at about one-half or one-third of the stroke; it is then exhausted by the pipe connecting the two cylinders, shown in fig. 106, from the

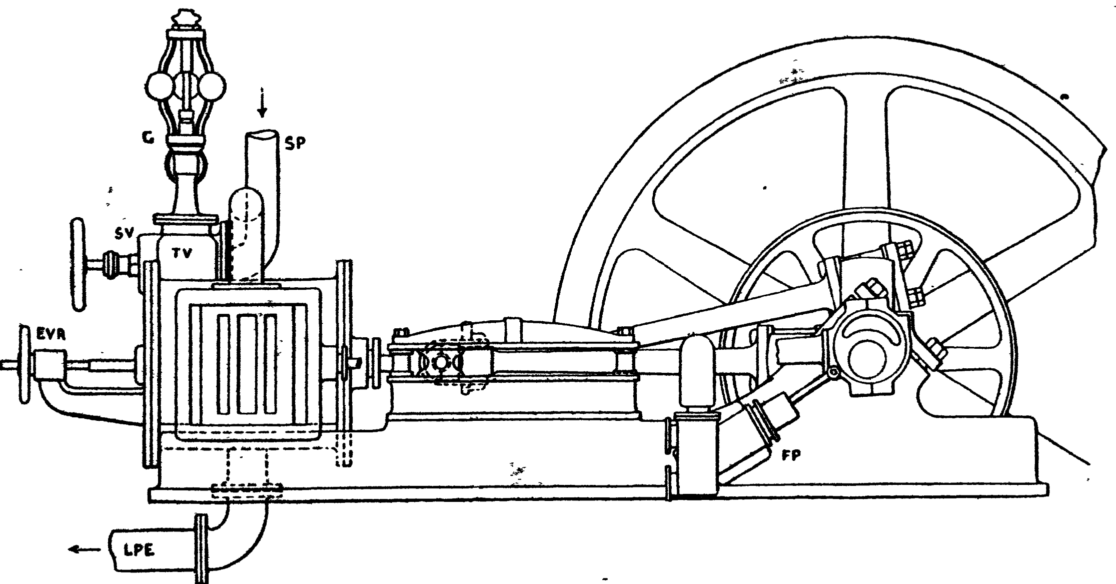


FIG. 104.

H P, high-pressure cylinder; L P, low-pressure cylinder; S P, steam pipe; T V, throttle valve; S V, stop valve; G, governor; L P E, low-pressure exhaust pipe; F P, feed pump; E V, expansion valve M, main slide valve; E V R, expansion valve regulator; E, eccentrics; P, pulley; F W, fly-wheel; C S, crank shaft.

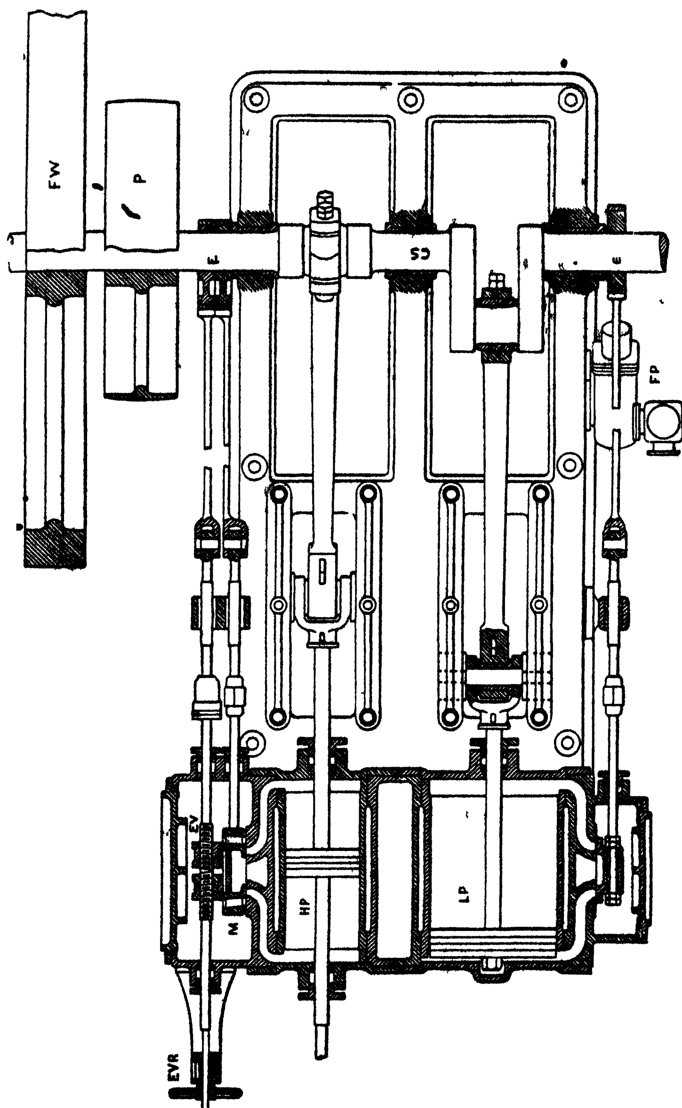


Fig. 105

high-pressure into the low-pressure cylinder, where it again does work by acting on the low-pressure piston. The steam is then exhausted, either into the air or into a condenser, by a pipe shown below the low-pressure cylinder.

In a two-cylinder compound engine the steam exhausted from the high-pressure cylinder into the low-pressure acts as forward pressure in the low, and as back pressure in the high,

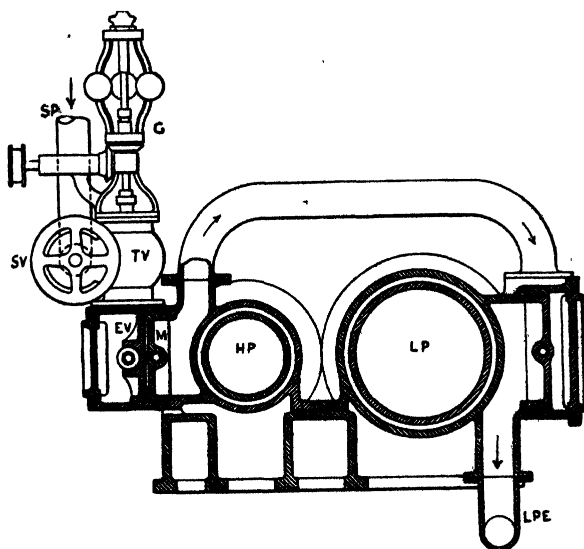


FIG. 106.

and the effective work done is due to the difference in area between the two pistons.

Thus, suppose steam admitted between two pistons of equal area fixed on a rod, as shown in fig. 107. Here it is evident that the pressures on the inner faces of the two pistons being equal and opposite, the pistons will not move in either direction from this cause, and the effective pressure transmitted to the piston rod R by P is quite independent of the pressure between the pistons.

But if the pressure acts on two pistons of *unequal* area, as

in fig. 108, the effective pressure transmitted by the pistons to the piston rod R is equal to the external pressure P on the small piston, plus the internal pressure on the difference of area between the large and small piston, less the back pressure *p* on the large piston; from which it will be seen that the greater the initial pressure P of the steam on the small or high-pressure piston, and the greater the pressure between the two pistons, and the less

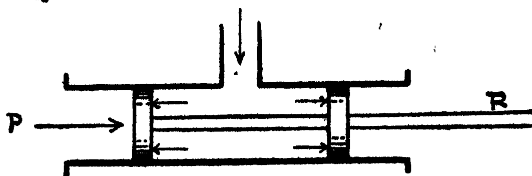


FIG. 107.

the back pressure *p* on the low-pressure piston, the greater the effective pressure transmitted.

The volume of the low-pressure cylinder of a compound engine required for a given power is the same as if the whole of the work to be done, and the whole of the expansions, were performed in that cylinder alone; and its size is therefore estimated

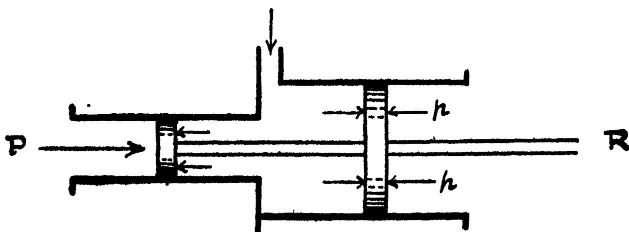


FIG. 108.

as for a single-cylinder engine, to exert the required power with the given initial pressure of steam of the high-pressure cylinder, admitted at once to the low-pressure cylinder and expanded down to the terminal pressure, the assumed point of cut-off being arranged to allow the same number of expansions as with the compound engine.

It will be evident that the volume of steam exhausted into the condenser at each stroke is the volume due to the capacity of the low-pressure cylinder; and, provided the terminal pressure is constant, the volume and weight exhausted at each stroke is constant, whether the steam was admitted at boiler pressure direct to the low-pressure cylinder and expanded in it down to the constant terminal pressure, or whether it has arrived there after passing through one, two, or more cylinders.

The size of the low-pressure cylinder having been determined, the remaining cylinder or cylinders are so proportioned as to equalise as much as possible the initial and mean stresses and the range of temperature.

The ratios of the volumes of the cylinders, or of the piston areas (all being of equal stroke), are as the squares of their diameters. Thus, if the low-pressure cylinder diameter be made twice that of the high-pressure, then their areas or volumes are as 1 : 4.

Or, again, if the cylinders of a triple expansion engine have their respective diameters in the proportion of 3, 5, and 8, then the areas of the successive pistons are to one another as $3^2 : 5^2 : 8^2 = 9 : 25 : 64 = 1 : 2{\cdot}78 : 7{\cdot}11$.

The number of expansions of the steam in any engine, whether simple or compound, $= \frac{\text{final volume}}{\text{initial volume}}$; and this is approximately equal to $\frac{\text{initial pressure}}{\text{terminal pressure}}$ where the pressures are expressed in lbs. per sq. in. *absolute*. Thus, neglecting the effect of clearance spaces, number of expansions = volume of low-pressure cylinder divided by volume of high-pressure cylinder to point of cut-off. For example, suppose that in a two-cylinder compound engine the ratio of the piston diameters is as 1 : 2, then the areas of the pistons and volumes of the cylinders are as 1 : 4. If, then, the steam were supplied to the high-pressure cylinder throughout the whole stroke and then exhausted into the low-pressure cylinder, the number of expansions would be

$$\frac{\text{final vol.}}{\text{initial vol.}} = \frac{\text{vol. of L. P. cylinder}}{\text{vol. of H. P. cylinder}} = 4.$$

But if the steam is cut off in the high-pressure cylinder at one third of the stroke, the number of expansions =

$$\frac{\text{final vol.}}{\text{initial vol.}} = \frac{\text{vol. of L. P. cylinder}}{\frac{1}{3}(\text{vol. of H.P. cylinder})} = \frac{4}{\frac{1}{3} \text{ of } 1} = 12.$$

Or, again, if the initial pressure of the steam in the high-pressure cylinder is 90 lbs. absolute, and the terminal pressure in the low-pressure cylinder is 10 lbs. absolute, then the number of expansions $= \frac{\text{initial pressure}}{\text{terminal pressure}} = \frac{90}{10} = 9.$

Suppose the ratio of the cylinder capacities is as 1 : 4, and we wish to expand the steam from 90 lbs. initial pressure absolute to 10 lbs. terminal pressure = 9 expansions. Here the steam must evidently be cut off at an early point in the stroke of the high-pressure cylinder, which point is found as follows:

$$\text{Let } R = \frac{\text{vol. of L. P. cylinder}}{\text{vol. of H. P. cylinder}} = 4.$$

Then the point of cut-off in the high-pressure cylinder =

$$\frac{R}{\text{number of expansions}} = \frac{4}{9} \text{ of the stroke.}$$

Example.—The ratio of the cylinder volumes of a two-cylinder compound engine are as 1 : 3; the initial pressure by boiler gauge is 75 lbs.; and the terminal pressure required is 10 lbs. absolute: find the point of cut off in the high-pressure cylinder. *Ans.* $\frac{1}{3}$ of the stroke.

A single cylinder is sufficient when steam is expanded not more than about 5 times; for a greater number of expansions the compound engine is more economical.

Thus, suppose the terminal pressure at which it is required to work is 10 lbs. absolute, using a condenser; then, if the pressure of the steam at our command is only, say, 40 lbs. by boiler gauge, that is 55 lbs. absolute, the number of expansions $= \frac{55}{10} = 5{\cdot}5$, or allowing for losses $= 5$, which would only require a single-cylinder engine. If, however, the pressure of steam at command is 90 lbs. per sq. in. by boiler gauge, or 105 lbs. absolute, the number of expansions $= \frac{105}{10} = 10{\cdot}5$, or

allowing for losses = 10, in which case a two-cylinder compound would be necessary.

Suppose the engine required is to be non-condensing, then with a terminal pressure of, say, 5 lbs. above the atmosphere, or 20 lbs. absolute, and a boiler pressure at command of 80 lbs., or 95 lbs. absolute, the number of expansions $= \frac{95}{20} = 4{\cdot}75$, or practically 4·5, in which case it would be unnecessary to use a compound engine.

The influence of clearance, and intermediate passages between the cylinders, will be considered presently.

CHAPTER XVI

TYPES OF COMPOUND ENGINES

COMPOUND engines may be roughly divided into two classes: (1) those in which the pistons of each cylinder commence the stroke simultaneously. In such engines the cranks are either at 0° or 180° apart. These engines are known as the 'Woolf' type. (2) Those in which the cranks are set at various angles other than 0° or 180°, and exhaust from one cylinder before the next cylinder is ready to receive it; in which case the steam is retained, for a portion of the stroke, in a chamber or receiver between the two cylinders. These are termed 'receiver' engines.

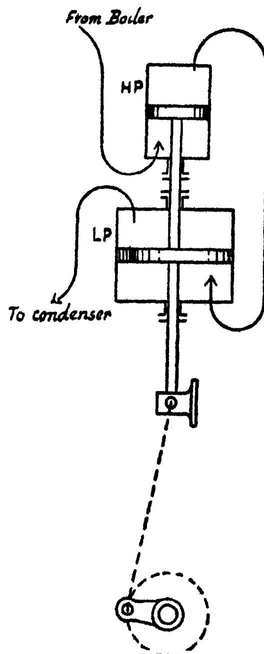


FIG. 109.

The following are the most common arrangements of cylinders and cranks of compound engines:

I. *The Tandem Compound Engine* with cylinders, as shown in fig. 109, the high-pressure cylinder being in line with the low-pressure cylinder, and the two pistons attached to the same piston rod. In the fig. 109, H P is the high-pressure cylinder and L P the low-pressure. Steam is conducted

the boiler direct to the high-pressure cylinder H. P., where it is admitted alternately at either end of the stroke, cut off at about one-half or one-third of the stroke, expanded nearly to the end of the stroke and then exhausted into the low-pressure cylinder L P, where it further expands, acting as back pressure on the high-pressure piston, and forward pressure on the low-pressure piston, and is finally exhausted into the air or a condenser.

The distribution of the steam in the cylinders of the tandem engine at various points in the stroke may be clearly followed by the aid of an ideal diagram. In order to simplify the diagram we will neglect the effect of clearance at the end of the cylinders, the connecting passages between the cylinders, the

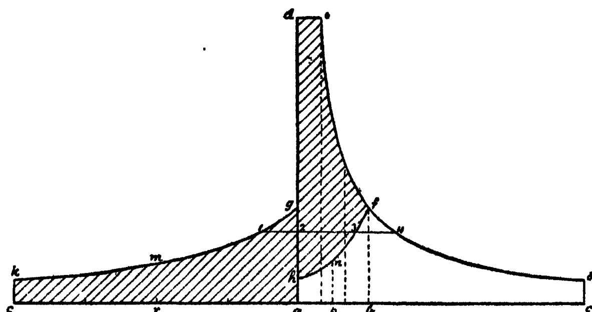


FIG. 110.

friction of the steam in the passages, compression, &c., and suppose the vacuum perfect.

In fig. 110, let the relative volumes of the high- and low-pressure cylinders be as 1 : 4, then make $a\,b = 1 =$ volume of high-pressure cylinder, and $a\,c = 4 =$ volume of low-pressure cylinder. From a set off $a\,d =$ the initial absolute pressure of steam in the high-pressure cylinder, the horizontal through a being the zero of pressures. Then, if the steam be admitted to the high-pressure cylinder for one-third of the stroke, $d\,e = \frac{1}{3}\,a\,b$ is the line of admission, and e is the point of cut-off, and $e\,f$ the curve of expansion to the end of the stroke of the high-pressure cylinder, the terminal pressure being $b\,f = \frac{1}{3}\,a\,d$. The steam is now exhausted into the low-pressure cylinder at an

initial pressure *a g* equal to the terminal pressure *b f*, and the two cylinders are now in direct communication. The volume of steam in the low-pressure cylinder increases as its piston moves forward, while at the same time the volume in the high-pressure cylinder decreases till its piston reaches the end of its stroke, and compresses the whole of the steam into the low-pressure cylinder. Here the volume of the steam is four times the volume of the high-pressure cylinder, and its pressure, therefore, falls to $ck = \frac{1}{4}$ *a g* or *b f*. During the time the cylinders are in communication, the pressure gradually decreases as the volume increases, but all the time it acts as back pressure on the high-pressure piston, and forward pressure on the low-pressure piston. The curve *g m k* represents the gradual fall of pressure as the volume of the low-pressure cylinder increases, and the curve *f n h* represents the decreasing back pressure on the high-pressure piston during the same period; $bf = ag$; $pn = rm$; and $ck = ah$. Then *d e f h* is the theoretical indicator diagram for the high-pressure cylinder, and *a g k c* for the low-pressure cylinder, and the areas of these figures represent the work done in each cylinder respectively. These two diagrams may be combined by drawing horizontal lines as 1, 2, 3, 4, and making 3, 4 = 1, 2, &c., and completing the curve to *s*.

The varying pressures and volumes throughout the stroke in compound engines, as in fig. 109, may be further illustrated by the aid of a numerical example, account being taken, in this instance, of the clearance spaces, and of the volume of the connecting passage or 'receiver' between the cylinders.

Thus, take the case of a compound tandem engine, as in fig. 109, of the following dimensions:

Volume of high-pressure cylinder . .	=5 cub. ft.
Clearance at each end of high-pressure	=·35 „
Volume of low-pressure cylinder . .	=20 „
Clearance at each end of low-pressure .	=1·2 „
Volume of connecting passage . .	=2·3 „

Cut-off at $\frac{1}{3}$ of the stroke in high-pressure cylinder; low-pressure cylinder takes steam to end of stroke.

Initial steam pressure = 100 lbs. per sq. in. absolute.

Then, the volume of steam admitted to high-pressure cylinder

$$= \frac{1}{3} \text{ volume of cylinder} + \text{clearance}$$
$$= \frac{1}{3} \text{ of } 5 + {\cdot}35 = 2{\cdot}02 \text{ cub. ft.}$$

The final volume of the steam is that contained by volume of low-pressure cylinder + clearance of low-pressure cylinder + clearance of high-pressure cylinder (omitting the steam in the intermediate chamber, which is a constant volume at the end of each stroke) $= 20 + 1{\cdot}2 + {\cdot}35 = 21{\cdot}55$ cub. ft.

$$\text{Then total ratio of expansion} = \frac{\text{final volume}}{\text{initial volume}}$$
$$= \frac{21{\cdot}55}{2{\cdot}02} = 10{\cdot}67,$$

and the terminal pressure of steam in the low-pressure cylinder

$$= 100 \times \frac{2{\cdot}02}{21{\cdot}55} = 9{\cdot}4 \text{ lbs. per sq. in.}$$

We may now trace the varying pressures of the steam in passing from the high-pressure cylinder, through the receiver, to the end of the stroke of the low-pressure cylinder.

Pressure at end of stroke of high-pressure cylinder

$$= 100 \times \frac{2{\cdot}02}{5{\cdot}35} = 37{\cdot}75 \text{ lbs. per sq. in.}$$

The steam is now exhausted at this pressure into the receiver.

If there were no intermediate chamber between the two cylinders—the steam passing from one immediately to the other—and no clearance, then the terminal pressure of the high-pressure cylinder, namely, 37·75 lbs., would be the initial pressure of the low-pressure cylinder (as in fig. 110); but when there is a connecting pipe—which answers the purpose of a receiver, and sometimes not an inconsiderable one—there is a fall or 'drop' in pressure owing to the increased volume now occupied by the steam. The receiver, however, is not empty when the high-pressure steam is exhausted into it, but it contains steam at a pressure, in the present case, equal to the terminal pressure of the low-pressure cylinder. When communi-

cation opens between the high-pressure cylinder and the receiver, we have, therefore, two volumes of steam at different pressures, namely, 5·35 cub. ft. at 37·75 lbs. pressure, in the high-pressure cylinder, and 2·3 cub. ft. in the receiver at 9·4 lbs. pressure (the terminal pressure in the low-pressure cylinder). The resulting pressure will therefore be equal to

$$\frac{(5{\cdot}35 \times 37{\cdot}75)+(2{\cdot}3 \times 9{\cdot}4)}{5{\cdot}35+2{\cdot}3}=29{\cdot}226 \text{ lbs.}$$

This is the pressure of the steam now occupying 7·65 cub. ft., namely, the volume of the high-pressure cylinder and receiver. On admission to the low-pressure cylinder, the volume is now increased by the clearance in the low-pressure cylinder, and it therefore now occupies $7{\cdot}65+1{\cdot}2=8{\cdot}85$ cub. ft.

Then assuming no back pressure against the low-pressure piston, the initial pressure on the low-pressure piston is, therefore, $29{\cdot}226 \times \frac{7{\cdot}65}{8{\cdot}85}=25{\cdot}26$ lbs. per sq. in.

To find the pressure at any intermediate point in the stroke, say $\frac{1}{4}$th; then the volume occupied by the steam will be: $\frac{3}{4}$ volume of high-pressure cylinder + clearance in high-pressure cylinder + volume of receiver + clearance in low-pressure cylinder + $\frac{1}{4}$ volume of low-pressure cylinder,

$$=\tfrac{3}{4} \text{ of } 5+{\cdot}35+2{\cdot}3+1{\cdot}2+\tfrac{1}{4} \text{ of } 20=12{\cdot}6 \text{ cub. ft.,}$$

and the pressure of the steam at this point acting as forward pressure on the low-pressure piston, and back pressure on the high-pressure piston, will be $29{\cdot}226 \times \frac{7{\cdot}65}{12{\cdot}6}=17{\cdot}74$ lbs. per sq. in.

The pressure at the end of the stroke may also be found from the same data; for volume of steam at end of stroke = volume of low-pressure cylinder + clearance of low-pressure cylinder + volume of receiver + clearance of high-pressure cylinder

$$=20+1{\cdot}2+2{\cdot}3+{\cdot}35=23{\cdot}85 \text{ cub. ft.,}$$

and its terminal pressure

$$=29{\cdot}226 \times \frac{7{\cdot}65}{23{\cdot}85}=9{\cdot}4 \text{ lbs. per sq in.,}$$

and this is the same result as we obtained before.

Communication is now opened with the condenser, and the pressure falls to that in the condenser.

The *range of temperature* in the cylinders may be followed for a special case by referring to a similar diagram to the previous one, having the pressures and temperatures marked upon it.

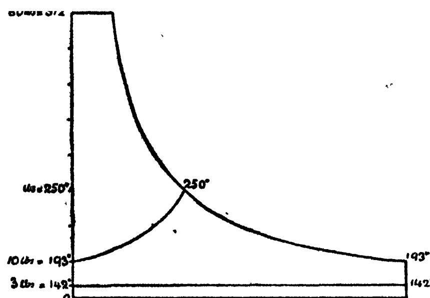


FIG. 111.

Thus, suppose steam at 80 lbs. absolute is expanded to 30 lbs. in the forward stroke of the high-pressure cylinder, and then to 10 lbs. in the low-pressure cylinder, after which it is exhausted into the condenser at 3 lbs. pressure. The temperatures due to these pressures are marked on the diagram, from which we may prepare the following table, showing the range of temperature in the cylinders of a compound Woolf engine as compared with a single-cylinder engine.

—	Forward Stroke	Exhaust Stroke	Total Range
Single-Cylinder Engine	312 − 142 = 170°	Open to condenser at 142°	170°
Woolf Engine H.P. Cylinder	312 − 250 = 62°	250 − 193 = 57°	119°
Woolf Engine L.P. Cylinder	250 − 142 = 108°	Open to condenser at 142°	108°

From the table it will be noticed that the range of temperature for the single-cylinder varies from the initial temperature of the steam to the temperature of the condenser—the two extremes—during the single forward stroke. The hot steam at the initial temperature of 312° enters the cylinder and meets with an internal metallic surface including cylinder cover, piston face, and steam port at a temperature of 142°.

(This assumes that there is no compression at the end of the stroke, the effect of which is to increase the temperature of the cylinder before admission.) But, further, during admission and expansion of steam in the forward stroke of the piston, the cylinder barrel on the other side of the piston is in communication with the condenser at 142°; and as the piston moves forward it brings more and more of the cold barrel into communication with the hot steam. It will not be surprising, therefore, that condensation of steam occurs in the cylinder under such conditions.

In the high-pressure cylinder of the Woolf engine, however, it will be seen that the range is very much less than in the single-cylinder engine. The steam at the initial temperature of 312° is admitted, when the temperature of the cylinder walls is 193°, or 51° warmer than with the single-cylinder engine. But, further, the cylinder barrel on the other side of the piston, the surface of which the piston in its forward movement brings more and more into communication with the initial steam, instead of being in communication with a condenser at 142°, contains steam at a temperature varying from 250° to 193°. There is, therefore, an evident gain in this arrangement over that of the single-cylinder engine.

The *variation of stress* on the mechanism of the engine is reduced by adopting the compound system. In the tandem arrangement, fig. 109, it will be evident that the maximum effective pressures of the steam upon the two pistons, the whole of which are transmitted through the working parts to the crank pin, occur at the same time. The sum of these pressures, however, is less than if the steam at the same initial pressure had been admitted direct from the boiler to a single cylinder of the diameter of the low-pressure cylinder. For if P=initial pressure per sq. ft. on high-pressure piston, and p=initial pressure per sq. ft. on low-pressure piston; and if the areas of the pistons are 1 sq. ft. and 4 sq ft. respectively, then $P \times 1$=pressure on high-pressure piston, and $p \times (4-1)$=effective pressure on low-pressure piston, and since p is less than P, then evidently $(P \times 1) + (p \times 3)$ is less than $P \times 4$.

Again, at the end of the stroke of the tandem engine,

the pressure on the crank pin is equal to the sum of the effective pressures on the two pistons; but in a single-cylinder engine working down to the same terminal pressure, the pressure would be that due to the pressure on the large piston only. We have, therefore, in the compound tandem engine the initial pressures less and the terminal pressures greater than in a single-cylinder engine of the same power, working through the same range of pressures. The effect may be represented by a diagram, fig. 112, which is approximately the figure that would be obtained if the pressures on the crank pin in the two cases were plotted by vertical ordinates measured from the zero line of pressure. From which we see that, though the *mean* pressures in the two cases may be the same, the range of stress is less in the compound engine. This limited range of stress, however, is not altogether an advantage, especially at high speeds.

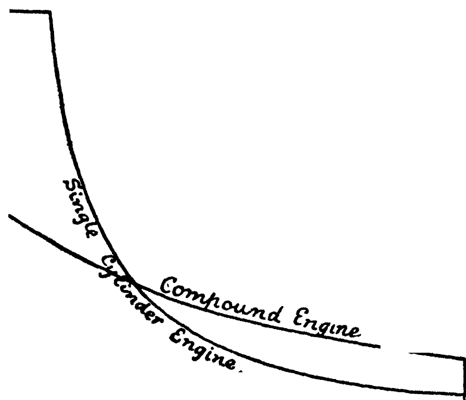


FIG. 112.

A further advantage in point of strength is gained in this engine over the single-cylinder engine, by using the steam of high initial pressure in a cylinder of small diameter.

II. *The Compound Engine, with the cylinders placed side by side*, and with the cranks at right angles, as shown at fig. 113.

In this engine the steam enters the high-pressure cylinder H P direct from the boiler, is cut off at about one-half or one-third of the stroke, and expands to the end of the stroke of the high-pressure piston, when it is exhausted into the receiver.

In practice it is usually found unnecessary to have a separate special chamber for a receiver, as the exhaust pipe of the high-

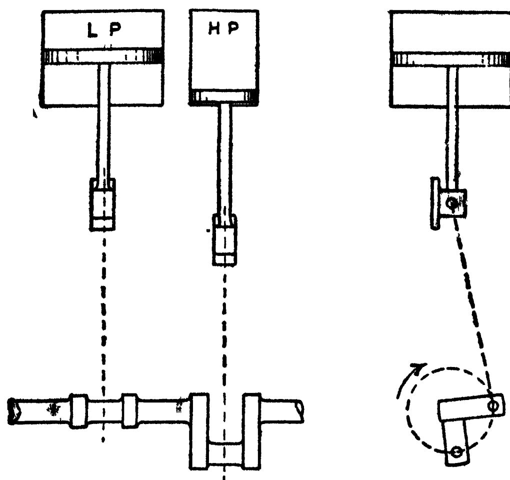


Fig. 113.

pressure cylinder and the valve chest of the low-pressure afford sufficient capacity for the purpose.

Suppose the steam is cut off in both the high- and low-

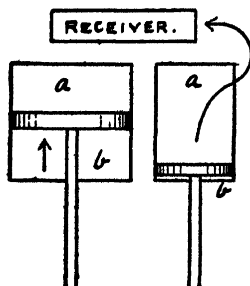


Fig. 114.

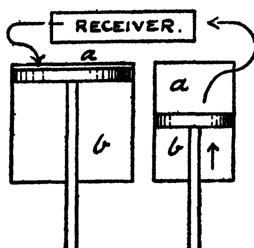


Fig. 115.

pressure cylinders at half stroke. Then, at the moment of exhaust from the high-pressure cylinder, the low-pressure piston

is only at half stroke (see fig. 114), and the low-pressure cylinder is therefore not yet ready to receive the steam.

The slide valve of the low-pressure cylinder, fig. 114, covers the port for admission of steam from the receiver to the side *a* of the low-pressure cylinder, because it is at present connected with the condenser; and as the cut-off in the low-pressure cylinder occurs at half stroke, the port will also be closed for admission of steam to the side *b*.

The position of the pistons is now as shown in fig. 114. The low-pressure piston proceeds to the end of its stroke, and the high-pressure piston (fig. 115) also returns from the bottom of the cylinder against the back pressure of its exhaust steam which fills the high-pressure cylinder and receiver, thereby reducing its volume and increasing its pressure. This proceeds till the high-pressure piston reaches its half stroke, by which time the low-pressure piston has reached the end of its stroke, and its steam port opens for admission of steam from the receiver to the end *a*, fig. 115. The confined steam in the receiver and high-pressure cylinder now expands, driving the low-pressure piston forward, and acting as back pressure on the high-pressure piston, and forward pressure on the low-pressure piston.

The initial pressure in the low-pressure cylinder (neglecting loss by friction in the passages) is equal to the pressure in the receiver when the high-pressure piston has reached the middle of its return stroke. This steam expands in the low-pressure cylinder till its piston reaches, say, half stroke, when its steam port closes. The supply of steam from the receiver being now cut off, it expands to the terminal pressure, and is exhausted into the condenser.

These operations may also be readily traced from an ideal indicator diagram. Suppose the steam cut off in both the high- and low-pressure cylinders at half stroke. In fig. 116 let the relative volumes of the high- and low-pressure cylinders be as 1 : 3. Make $ab = 1$ = volume of high-pressure cylinder, and $ac = 3$ = volume of low-pressure cylinder. From *a* set off ad = the initial absolute pressure of steam in the high-pressure cylinder, the horizontal through *a* being the zero of pressures.

Then, if the steam be admitted to the high-pressure cylinder for one-half the stroke, $de = \frac{1}{2} ab$ is the line of admission, e is the point of cut-off, and ef the curve of expansion to the end of the stroke of the high-pressure cylinder, the terminal pressure being $bf = \frac{1}{2} ad$. Communication is now opened with the receiver, and the pressure falls to g, the pressure bg depending on the volume of the receiver and on the pressure of the steam in it. But there is as yet no admission to the low pressure cylinder till another half stroke has been made (as shown by fig. 114). The diagram of work done by the high-pressure piston will therefore show an increasing back-pressure curve gt as that piston returns, till it reaches half stroke, when the low-

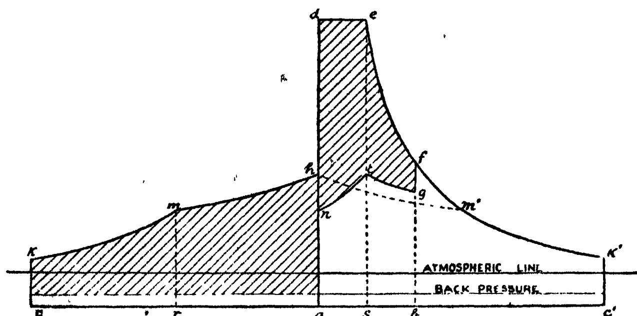


FIG. 116.

pressure steam port opens and admits steam at the initial pressure $ah = st$. The pressure now falls by expansion of the steam behind the low-pressure piston, the terminal pressure an in the high-pressure cylinder being equal to the pressure rm in the low-pressure cylinder at half stroke. Cut-off now takes place in the low-pressure cylinder, and the steam expands behind the piston to $ck = \frac{1}{6} ad = \frac{1}{3} bf$, at which point it escapes to the condenser, when the pressure falls to the line of back pressure.

If the cut-off in the low-pressure cylinder occurs later than half stroke, which it frequently does, there will be a momentary rise of pressure in the middle of the low-pressure diagram, due to the augmented pressure in the receiver from the high-

pressure exhaust; there will also be a corresponding fall of back pressure on the high-pressure piston. In practice, the changes indicated do not occur so as to produce sharp corners as shown on the ideal diagram. All the corners would be rounded, and the line *g t n*, for example, would be a gentle curve.

We may further illustrate this case by a numerical example. Take an engine as in fig. 113, with cranks at right angles, cut-off at half stroke in each cylinder.

Volume of high-pressure cylinder . .	=5 cub. ft.
Volume of low-pressure cylinder . .	=15 "
Volume of receiver	=8·5 "

Initial pressure of steam=120 lbs. per sq in. absolute.

Then, omitting the effect of clearance, steam is admitted to the high-pressure cylinder at an initial pressure of 120 lbs=*a d*, cut-off at half stroke=*d e*, and expanded to end of stroke=*e f*, where the terminal pressure *b f*=60 lbs.

The total rate of expansion$=\frac{\text{final volume}}{\text{initial volume}}=\frac{ac}{as}=\frac{15}{2\cdot5}=6$;

and the terminal pressure *c k* in low-pressure cylinder$=\frac{120}{6}$ =20 lbs. per sq. in.

The steam in the low-pressure cylinder is expanded twice in that cylinder, therefore the pressure *r m* at half stroke$=c\,k\times2$ $=20\times2+40$ lbs. per sq. in.; and this is also the pressure *a n* in the receiver at the point of cut-off.

The high-pressure cylinder exhaust now opens to the receiver, and we have two volumes of steam at different pressures, combining to fill the space, namely: volume of high-pressure cylinder at pressure *b f*, and volume of receiver at pressure *a n*, making a total volume of $5+8\cdot5=13\cdot5$ cub. ft., at a resultant pressure *b g* $=\frac{(5\times60)+(8\cdot5\times40)}{5+8\cdot5}=47\cdot4$ lbs. per sq. in., showing a 'drop' or fall in pressure $fg=60-47\cdot4=12\cdot6$ lbs. per sq. in.

This steam, however, cannot yet be admitted to the low-pressure cylinder because the steam port of that cylinder remains closed for another half stroke; hence the enclosed steam is

compressed behind the high-pressure piston during one half of its exhaust stroke, as shown by the line *g t*. The volume of the steam has by this time been reduced to $13·5-2·5=11$ cub. ft., and its pressure *s t* has been increased to $47·4 \times \frac{13·5}{11}=58·2$ lbs.

The steam port of the low-pressure cylinder is now opened, and steam at 58·2 lbs. initial pressure (*a h*) acts against the low-pressure piston. Here it is driven before the high-pressure piston and drives the low-pressure piston before it until the latter reaches half stroke, when the volume of the steam is now equal to $\frac{1}{2}$ volume of low-pressure cylinder + volume of receiver $=7·5+8·5=16$ cub. ft., and its pressure has fallen to $47·4 \times \frac{13·5}{16}=40$ lbs. per sq. in. $=r\,m$.

The supply of steam to the low-pressure cylinder is now cut off from the receiver, and we have in that cylinder a volume of steam equal to 7·5 cub. ft., which expands to end of stroke occupying 15 cub. ft., and having a terminal pressure of $40 \times \frac{7·5}{15}=20$ lbs. per sq. in. $=c\,k$ as obtained before.

Communication is now opened with the condenser, and the pressure *c k* falls to the line of back pressure.

The *range of temperature* in the high-pressure cylinder of the receiver compound engine with the cranks at right angles is less than in the Woolf engine. Here the total range in the high-pressure cylinder varies theoretically from the initial temperature of the steam to the temperature in the receiver when the low-pressure piston is in the middle of its stroke. In practice the range is not greater than that given by the difference between the initial temperatures in the two cylinders. The advantage of the distribution of the stresses between two cranks at right angles has been explained at p. 104.

Triple and quadruple expansion engines—namely, those in which the steam is expanded in three or four cylinders respectively—are the necessary outcome of increased pressures of steam; for, since the terminal pressure is about constant, increased pressures involve an increased number of expansions. And in order to prevent undue range of stress and temperature,

three and even four cylinders are now employed. Thus the same reasons which led to the rejection of the single-cylinder engine in favour of the two-cylinder compound, have now led to the rejection of the two-cylinder engine (at least, in marine work), and the adoption of the triple compound, and in some cases the quadruple compound, in its stead.

The economy of fuel which resulted from the introduction of high-pressure steam, and the compound engine with surface condensation for steamships, was very remarkable, as may be seen from the following table:—

Year.	Pressure of steam by boiler gauge per sq. in.	Consumption of coal per I.H.P. per hour.
1830	2 to 3 lbs.	9·0 lbs.
1840	8 "	5·5 "
1850	14 "	4·0 "
1860	30 "	3·0 "
1870	40 to 50 "	2·6 "
1880	70 to 80 "	2·2 "
1886	150 to 160 "	1·5 "
1889	—	1·4 "

Between 1860 and 1870, when the pressure of steam used for marine engines was about 30 lbs. by boiler gauge, and the steam expanded in a single cylinder, the amount of coal consumed by the best engines was about 4 lbs. per I.H.P. per hour.

On the introduction of the compound engine, the consumption fell to a little over 2 lbs. per I.H.P. per hour. The triple expansion engine has reduced this to as low as 1·4 lbs. per I.H.P. per hour, and the quadruple expansion engine still further reduces the consumption by about 10 per cent. To appreciate the significance of so apparently small a gain as $\frac{1}{4}$ lb. of coal per I.H.P. per hour, we will take an example:

Suppose a vessel of 6,000 I.H.P. steams from London to Melbourne and back in eighty-four days, find the saving on such a trip.

Gain per I.H.P. per hour $= \frac{1}{4}$ lb. of coal.

" " per day $= \frac{1}{4} \times 24$ lbs. of coal.

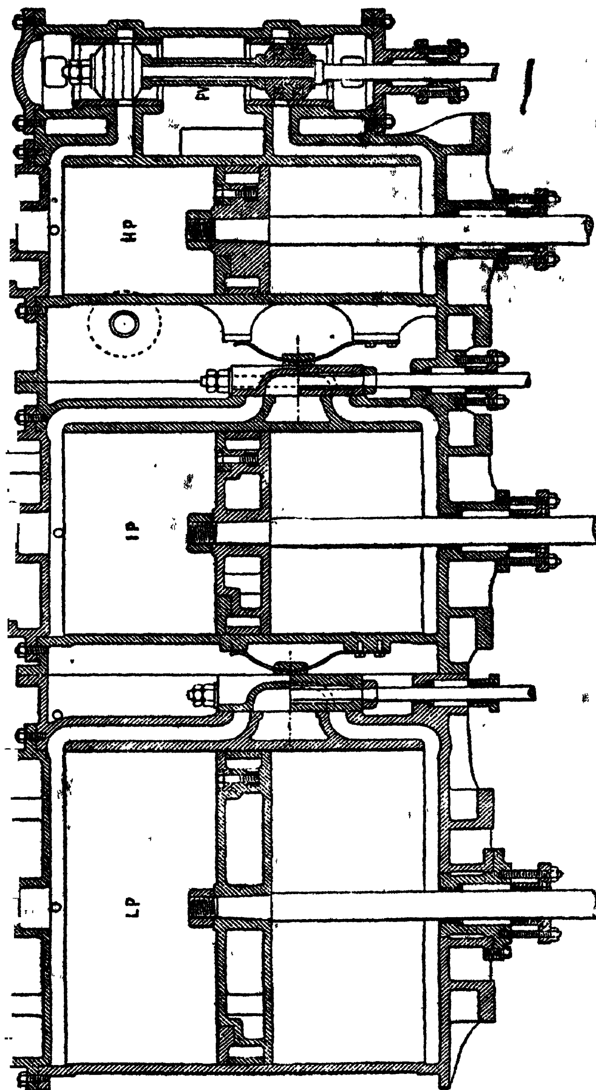


Fig. 117.

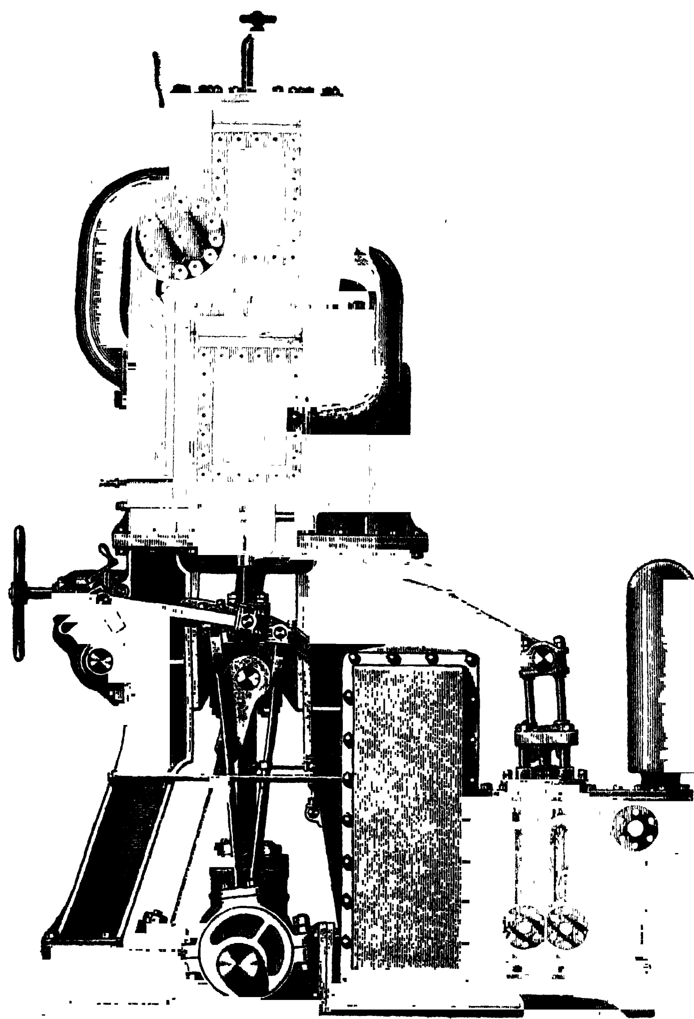


Fig. 118.

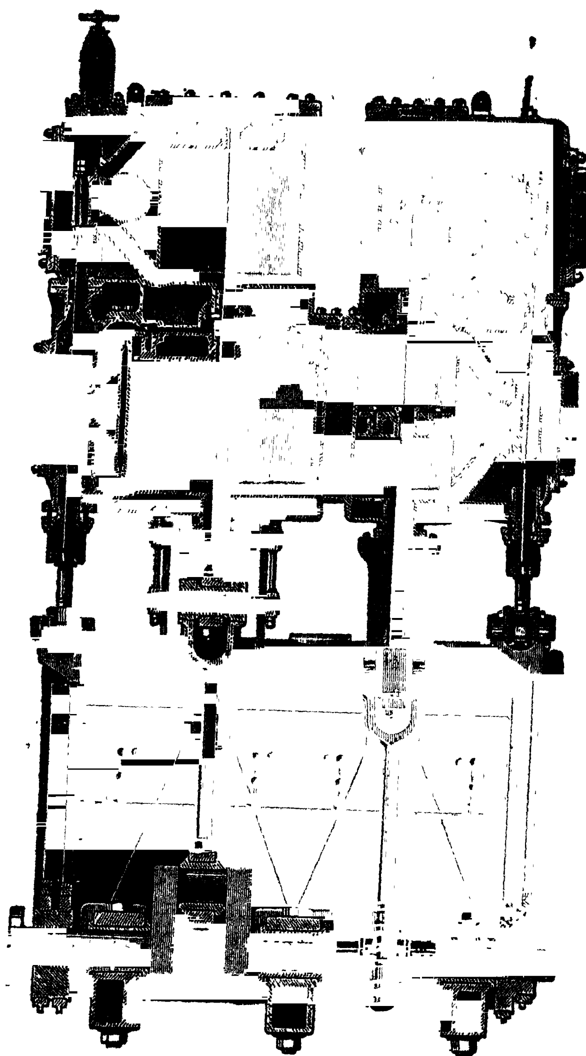


FIG. 119.

Gain per I.H.P. per 84 days $=\frac{1}{4}\times 24\times 84$ lbs. of coal.
,, per 6,000 I.H.P. per 84 days $=\frac{1}{4}\times 24\times 84\times 6{,}000$ lbs.
$=3{,}024{,}000$ lbs.
$=1{,}350$ tons.

The principles which govern the construction of triple and quadruple expansion engines are merely an extension of those already considered.

The diagram, fig. 117, is a section through the cylinders of a set of triple expansion paddle engines made by Messrs. Bow, McLachlan and Co., of Paisley.

The cylinders of these engines are: High-pressure cylinder, 16 in.; intermediate, 25 in.; low-pressure, 39 in. diameter, respectively, having a stroke of 36 in. The cranks are set at an angle of 120° relative to each other. The high-pressure cylinder is fitted with a piston valve, the exhaust steam being led round by a passage to the casing of intermediate cylinder, which cylinder is fitted with an ordinary flat-faced slide valve; from this cylinder the steam is led round by a passage formed on intermediate cylinder to low-pressure cylinder, which is fitted with a flat-faced slide valve. The exhaust steam is led from thence to the condenser, which is arranged under the crank shaft.

An enlarged drawing of the piston valve is shown in fig. 68.

Figs. 118 and 119 show a sectional view of a set of quadruple expansion engines made by Messrs. Fleming and Ferguson, of Paisley. Fig. 119 gives a section through all four cylinders, showing the slide valves, steam ports, and the construction of the pistons and stuffing boxes. The steam enters the smallest or high-pressure cylinder only, direct from the boiler; and it is then successively expanded to the second, third, and fourth cylinders (in the order of their diameters) by means of the pipes shown on the other view; and finally into the condenser, shown as a rectangular box, forming part of the engine framing.

The diameters of the cylinders are 10½ ins., 14 ins., 20 ins., and 28 ins. respectively; the stroke is 20 ins.; and the indicated horse-power 360. The whole of the low-pressure cylinder, and the bottom of the third cylinder, are jacketed. The upper cylinders form the covers for the lower. The lower cylinders have hand holes in front, to allow of their pistons being sighted,

and the junk ring pins felt without disturbing the upper cylinders. The packing between the cylinders is of the self-adjusting spring metallic type, and will last for years without attention. The crank shaft is 5½ ins. diameter, with cranks at right angles; it is forged from the best wrought-iron scrap, and has three bearings, 10 ins. in length. The condenser has 390 sq. ft. of cooling surface.

Air pump . .	12 ins.	diameter,	12 ins.	stroke.
Circulating pump .	7 ins.	"	12 ins.	"
Feed pump . .	2¾ ins.	"	12 ins.	"
Bilge pump . .	2¾ ins.	"	12 ins.	"

The heating surface in the boiler is 752 sq. ft., and the grate surface 27 sq. ft.

Working pressure of steam in boiler, 200 lbs. per sq. in.

CHAPTER XVII

BOILERS

THE vessel in which the steam is generated is called the boiler.

In the early days of boiler construction, the pressures used were not higher than 3 or 4 lbs. on the square inch; and boilers were then constructed without regard to suitability of form to resist internal pressure. But, as steam pressures began to increase, increased attention to this point became necessary. To-day 150 lbs. on the square inch is not uncommon, and to carry this safely the strongest possible form of boiler must be adopted.

The sphere is the strongest form of vessel to resist internal pressure, but there are many practical reasons which prevent its being used for the purpose. Next to the sphere the cylindrical form is the simplest and strongest, and it is now universally adopted.

Resistance of cylindrical vessels when subjected to internal pressure.—Let fig. 120 represent a thin cylindrical vessel subjected to internal pressure. Let p = internal pressure per square inch; d = diameter of cylinder; t = thickness of plate; and l = length of cylinder.

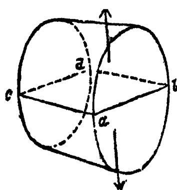


FIG. 120.

It is evident that the internal pressure p is acting radially from the centre on every part of the internal circumference of the shell; but if these forces be resolved into components parallel and perpendicular to a given plane, the resultant forces, tending to separate the cylinder into two parts through a plane $a\,b\,c\,d$, can be shown to be equal to $p \times d \times l$.

The area of the material to resist this tendency to burst along the lines ac and $bd = (ac + bd)\ t = 2\ l \times t$. Hence the stress ($s$) per square inch on the plate may be expressed thus:

$$s = \frac{\text{load}}{\text{area}} = \frac{p\,d\,l}{2\,t\,l} = \frac{p\,d}{2\,t}. \quad . \quad . \quad . \quad (1)$$

From which we learn that the stress on the material increases as the pressure or the diameter increases; and the stress per sq. in. on the material decreases as the thickness of the material is increased. In practice the section would be taken through the weakest part, which, in a new boiler, is through the rivet holes. The strength of a single riveted joint is taken as 56 per cent. that of the solid plate, and of a double riveted joint 70 per cent.

Again, to find the pressure tending to tear the boiler in two in a plane perpendicular to the axis; in other words, tending to blow the end off.

$$\text{Area of end} = d^2 \times \cdot 7854.$$
$$\text{Total pressure on end} = (d^2 \times \cdot 7854)\,p.$$

The area of the material to resist this tendency (neglecting deductions for rivet holes) = circumference of shell × thickness of plate $= d \times 3 \cdot 1416 \times t$. Hence the stress ($s$) per square inch on the plate may be expressed thus:

$$s = \frac{\text{load}}{\text{area}} = \frac{d^2 \times \cdot 7854 \times p}{d \times 3 \cdot 1416 \times t} = \frac{p\,d}{4\,t}. \quad . \quad . \quad . \quad (2)$$

Comparing this result with that in (1) above, we see that the stress is only half as great in the latter case; in other words, theoretically the plate is twice as likely to give way in the direction of the length of the boiler as circumferentially. For this reason the longitudinal joints are made stronger than the circumferential by the addition of an extra row of rivets.

Descriptions of Boilers

Boilers may be divided into three classes: stationary, locomotive, and marine.

Stationary Boilers

The Cornish boiler.—This form of boiler was first adopted by Trevithick, the Cornish engineer, at the time of the intro-

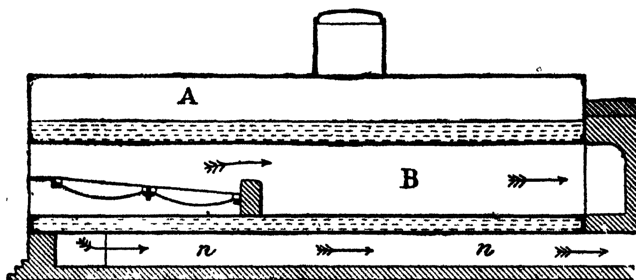


Fig. 121.

duction of high-pressure steam to the early Cornish engine, and it is still much used.

It consists of a cylindrical shell A, with flat ends, through which passes a smaller tube B containing the furnace, as shown in fig. 121. The products of combustion pass from the fire-grate forward over the brickwork bridge to the end of the furnace tube; they then return by the two side flues *m m'* to the front end of the boiler, and again pass to the back end by a flue *n n* along the bottom of the boiler to the chimney. Fig. 122 shows a transverse section of the boiler and flues.

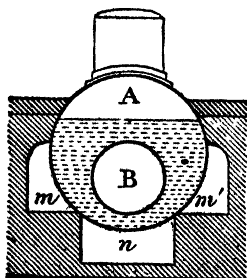


Fig. 122.

One advantage possessed by this type of boiler is that the sediment contained in the water falls to the bottom, where the plates are not brought into contact with the hottest portion of the furnace gases. The reason for carrying the products of combustion first through the side flues, and lastly through the bottom flue, will now be evident; for the gases, having parted with much of their heat by the time they reach the bottom flue, are less liable

to unduly heat the plates in the bottom of the boiler, where sediment may have collected.

Water tubes are often fitted to Cornish and Lancashire boilers. Their shape and position will be understood from the diagram, fig. 123. Holes are cut opposite each other in the furnace tube, and the joints made good by riveting the flanges of the water tube round the hole. Water can thus flow freely through the tube. They pass right across the furnace beyond the furnace bars, so that the flame and hot gases have a considerably increased surface to act upon. Besides increasing the heating surface, these tubes improve the circulation of the water, and act as a stay to the furnace tube. They are not an unmixed good, however, for they cool the furnace gases and retard combustion.

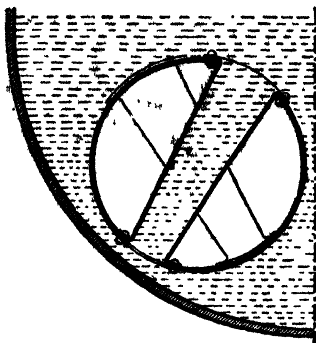


FIG. 123.

The Lancashire boiler differs from the Cornish boiler in having two internal furnace tubes instead of one. The separate furnaces are intended to be fired alternately, so that the mixture of smoke and unburnt gases from the newly-fired furnace may be consumed in the flues by the aid of the high temperature of the gases from the bright fire of the other furnace.

The following figures (figs. 124, 125 and 126) illustrate the construction of a Lancashire boiler. Fig. 124 shows a longitudinal section.

The furnace door, P, opens to the furnace where the fuel is supported on two or three successive lengths of fire-bars, underneath which is the ash-pit. At the back end of the furnace is a low brickwork bridge. Besides limiting the length of the fire-grate, the bridge causes the flame to rise against the upper surface of the tube. The fire-bars are supported on bearers. The front bearer, which is a cast-iron plate, is called the *dead plate.* Beyond the furnace are shown the Galloway tubes *a a a*,

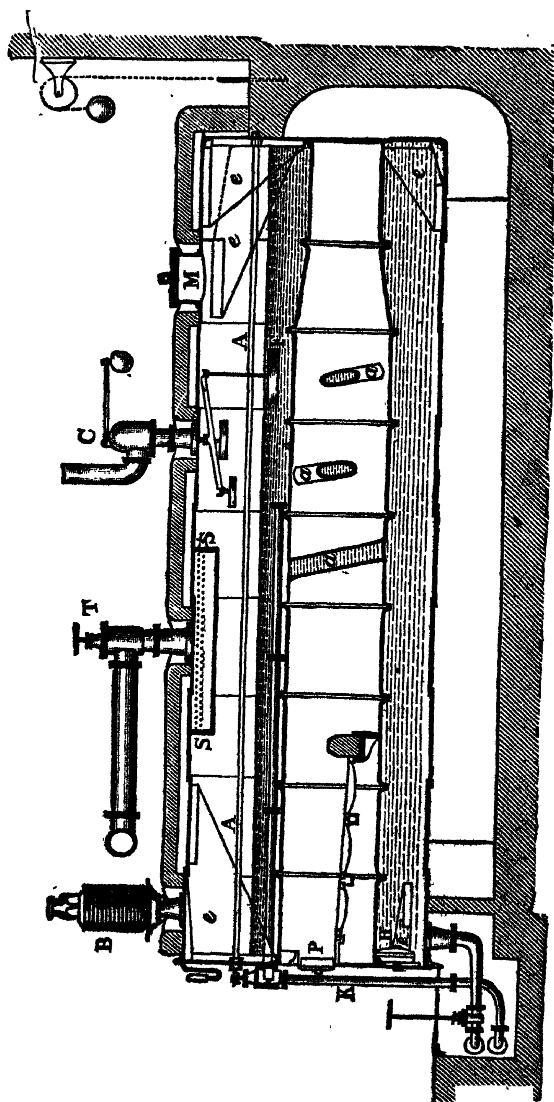


Fig. 124.

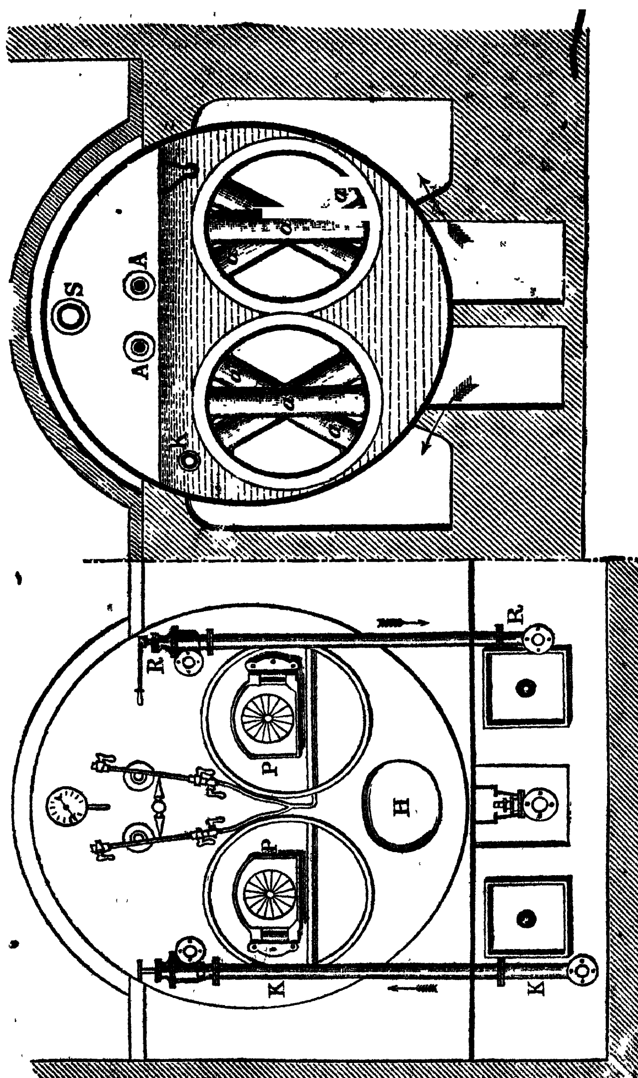
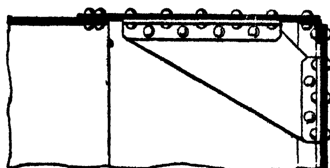


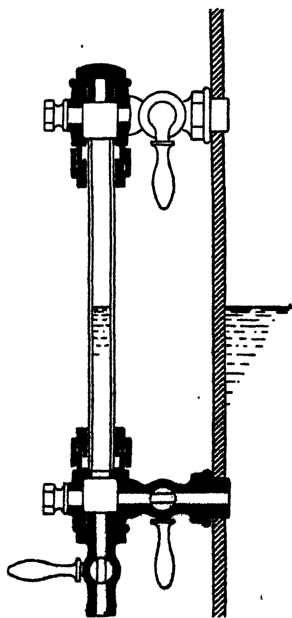
Fig. 126.

Fig. 125.

seen also in fig. 126. In the Lancashire boiler the furnace gases pass to the end of the furnace tube, and then by the flue underneath the boiler to the front, where it divides and again passes by side flues (see fig. 126) to the back end of the boiler and up the chimney. The flat ends of the boilers are prevented from bulging by the furnace tubes and by longitudinal stays, A A, also by gusset stays *e e*, shown in fig. 124 and enlarged in fig. 127.


Fig. 127.

The level of the water is shown, and the space above this is occupied by the steam. The steam is collected in the pipe S, which is perforated with holes all along the top so as to admit the steam, and at the same time prevent water spray from passing to the engine with the steam. On opening the stop valve T (see also fig. 139), the steam passes by the steam pipe to the engines. Two safety valves are shown, one a *dead-weight* safety valve B (see also fig. 138), and the other a lever safety valve C.


Fig. 128.

The float F is balanced so as to float on the surface of the water. Should the water fall below a safe level, the float F, which falls with the water, causes the valve to open by means of levers, and allows steam to escape, giving warning of shortness of water.

A manhole M is shown, by which access is obtained to the interior of the boiler for cleaning and inspection or repairs.

A mudhole H is also required for cleaning out the boiler, and removing the sediment which accumulates.

A blow-off cock and pipe is shown in the bottom of the boiler at the front end. On the front of the boiler, fig. 125, is shown a pressure gauge with a finger indicating the pressure of the steam in the boiler above the atmosphere ; two water-gauge glasses showing the height of the level of the water in the

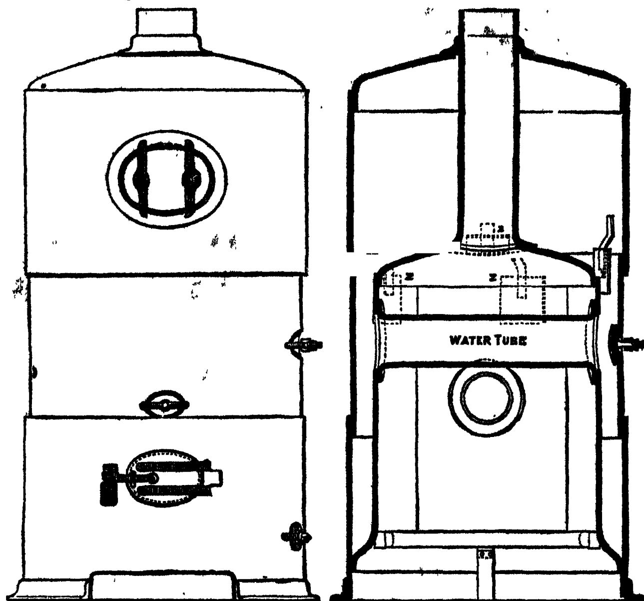


FIG. 129.[1]

boiler (see enlarged view, fig. 128) ; the furnace doors P ; the feed-pipe K, which is shown extending some distance into the boiler in fig. 124 ; and the scum cock R for blowing off the scum which accumulates on the surface of the water.

VERTICAL BOILERS

The illustration, fig. 129, shows the construction of a vertical boiler. These boilers are used for small powers, and where

[1] From *The Marine Engine*, by Mr. R. Sennett (Longmans).

space is limited. The internal fire-box is frequently made slightly tapering towards the top to allow of the ready passage of the steam to the surface. The bottom of the fire-box is attached to the bottom of the outer shell by being flanged out as shown, or by means of a solid wrought-iron ring, as shown in the locomotive boiler, fig. 132, the rivets passing right through the plates and solid ring. The water tubes pass across the internal fire-box, and increase the heating surface as well as improve the circulation, though they cool the furnace gases. The plate forming the passage leading from the top of the fire-box to the chimney—called the *uptake*—is frequently protected either with fireclay or with a cast-iron liner.

THE MARINE BOILER

Marine boilers, which are now usually constructed to carry steam at pressures up to 150 or 160 lbs. per square inch, are cylindrical tubular boilers, short as compared with their diameter, as illustrated by the accompanying diagram (fig. 130).

Description of the figure.—The boiler is of the cylindrical, multitubular type, fired from one end, with three furnaces. The products of combustion in the furnaces are carried forward by the draught into the combustion chambers C C, and thence through the tubes in the direction of the arrow to the front of the boiler, whence they pass up the funnel.

The *outside shell* is 12 ft. $1\frac{5}{8}$ in. extreme diameter, and 9 ft. $5\frac{13}{16}$ ins. extreme length. The plates are of steel, $\frac{13}{16}$ in. thick, in three rings united together circumferentially by double-riveted lap joints. The longitudinal seams are treble-riveted. The end plates are made in three pieces, and are joined together by double-riveted lap joints, and flanged to meet the shell and the furnace flues.

The *furnaces* are 3 ft. inside diameter, constructed of Fox's corrugated steel plates $\frac{1}{2}$ in. thick. They are flanged at the back end, and riveted to the combustion chambers.

The *combustion chambers* are flat on the top, and are supported by wrought-iron girder stays. The back and sides of these chambers are stayed with $1\frac{3}{8}$-in. screwed stays, fitted with nuts on both ends.

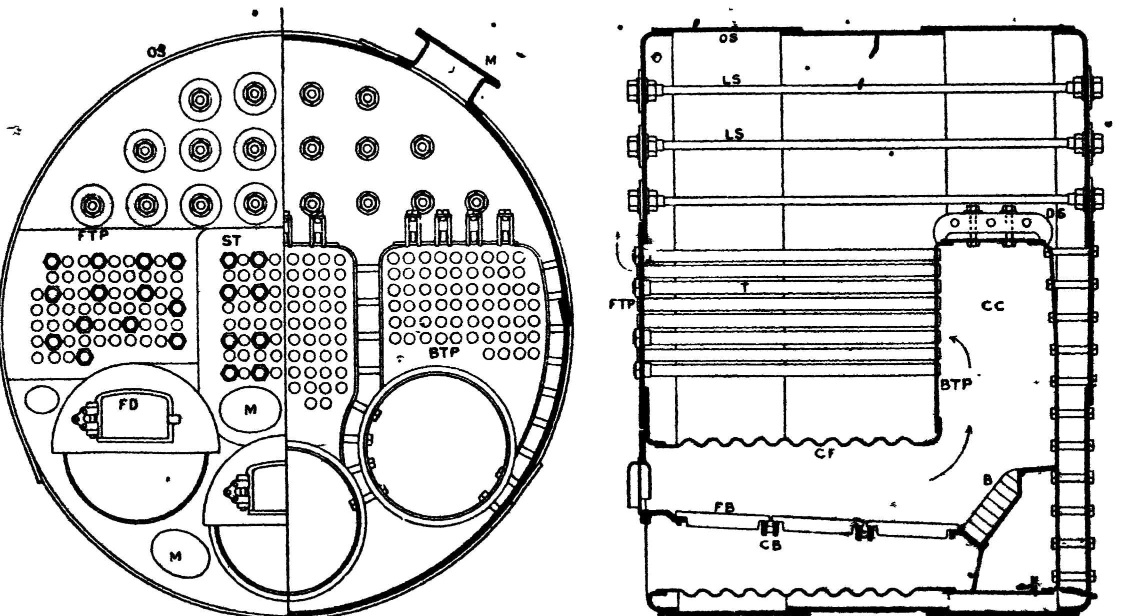


FIG. 130.

O S, outer shell; C F, corrugated flues; F B, fire-bars; B, brickwork bridge; C C, combustion chamber; B T P, back tube plate; F T P, front tube plate; L S, longitudinal stays; D S, dog stays; S T, stay tubes; F D, fire-door; M, manhole; C B, cast-iron bearer.

The boiler contains 200 tubes, 3 ins. diameter outside, of which 42 are stay tubes. The stay tubes are of wrought iron, $\frac{5}{16}$ in. thick, and screwed into the plates with nuts on the front ends. The remainder of the tubes are of brass.

Longitudinal stays, $1\frac{7}{8}$ in. diameter, steel, pass through the steam space from end to end, and support the front and back plates of shell.

Fire-grate area=area of grate × number of furnaces=

$$(3\times6)\times3=54 \text{ sq. ft.}$$

The heating surface.—The effective heating surface of a marine boiler is obtained by finding the sum of the following areas :

1. Area of furnace above level of fire-bars.

2. Area of sides and crown of combustion chamber above level of bridge.

3. Area of back tube plate, less area of holes for tubes.

4. Area of surface of tubes, namely, the area obtained by multiplying the external circumference by the length *between* the tube plates. The area of the front tube plate is omitted.

The length of furnace should not exceed 6 ft., otherwise it becomes difficult to stoke. The fire-doors are made of three pieces of plate placed about $2\frac{1}{2}$ ins. apart, the two inner ones being perforated. It will be noticed that the back of the combustion chamber slopes a little inwards towards the top. This enables the steam to rise more freely.

The space allowed between the tubes is 1 in., and the tubes are arranged in vertical rows to allow of the boiler being properly cleaned internally.

Manholes are placed on the top and front of the boiler, to get at the upper and lower parts of the furnaces for cleaning and repairing. The furnace bars are of wrought iron, and in three lengths, sloping towards the bridge $\frac{3}{4}$ in. per foot. Distance between bars $\frac{1}{2}$ in., maintained by widened ends of bars.

Steam room.—It is important to have as large a reservoir of steam as possible above the level of the water in the boiler, to prevent too great fluctuations of pressure. The water level should be at least 7 ins. above the top row of tubes.

To find the cubic contents of the steam space : Find the

area of the segment of the circle occupied by the steam, and multiply by the internal length of the boiler; and from this subtract the contents of the stays which occupy part of the steam space.

To find the area of the segment of a circle (fig. 131): Area of whole circle $\times \frac{\text{angle } a c b}{360}$ — area of triangle $a b c$.

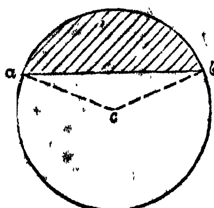


Fig. 131.

To give the front and back plates of shell the necessary stiffness, large circular plate washers, 10 ins. diameter, are riveted on to outside of plates.

The maximum stress allowed on these stays is 8,000 lbs. per square inch for stays under $1\frac{1}{2}$ in. diameter, and 9,000 lbs. for stays over $1\frac{1}{2}$ in.

The Locomotive Boiler

The following diagram (fig. 132) is a longitudinal section of the locomotive boiler. The fire-box F B, or furnace, is of rectangular section, and is made of copper, stayed by means of screwed and riveted copper stays, $\frac{7}{8}$ in. in diameter and 4 ins. apart, to the outer shell of the boiler.

The crown plate of the fire-box being flat requires to be very efficiently stayed, and for this purpose girder stays called fire-box roof stays are mostly used, as shown in the figure. These stays are now being made of cast steel for locomotives. They rest at the two ends on the vertical plates of the fire-box, and sustain the pressure on the fire-box crown by a series of bolts passing through the plate and girder stay, secured by nuts and washers. Fig. 133 is a plan and elevation of a wrought-iron roof stay.

Another method adopted in locomotive types of marine boilers for staying the flat crown of the fire-box to the circular shell plate is shown in fig. 134—namely, by wrought-iron vertical bar stays secured by nuts and washers to the fire-box and with a fork end and pin to angle-iron pieces riveted to the outer shell.

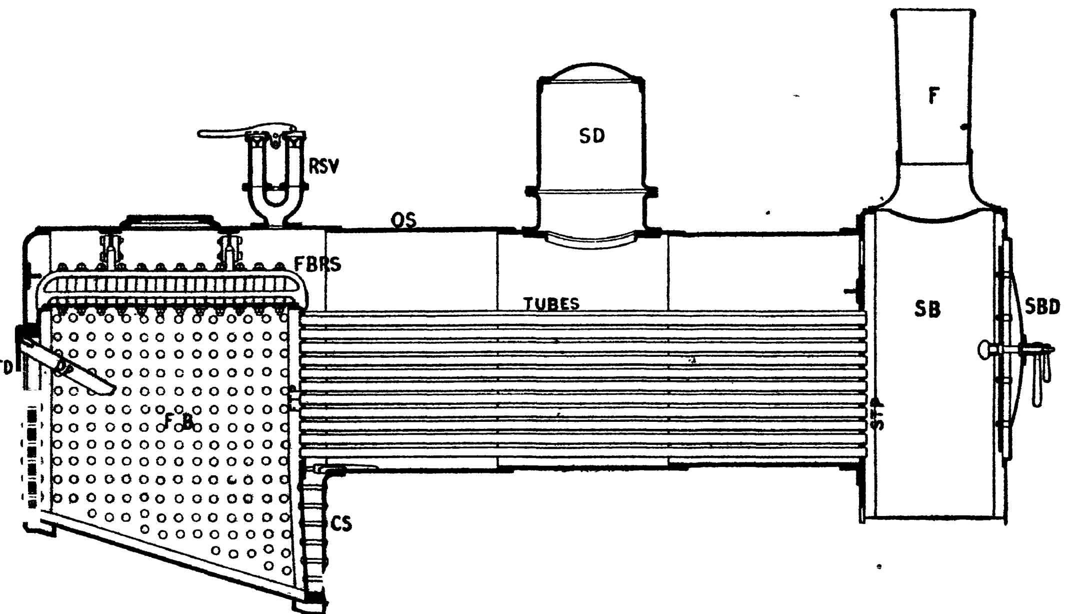


FIG. 132.

F B, fire-box; F D, fire-door; D P, deflector plate; F T P, fire-box tube plate; F B R S, fire-box roof stays; S T P, smoke-box tube plate
S B, smoke-box; S B D, smoke-box door; S D, steam dome; O S, outer shell; R S V, Ramsbottom safety valve; F, funnel or chimney.

The barrel of the boiler contains the tubes through which the products of combustion pass. The advantage of the tubes is the large amount of heating surface they expose to the heated gases. If the tubes are placed too closely together the steam generated round the tubes cannot freely escape; and as steam cannot absorb the heat so readily as water, the surface of the tube is liable to be overheated and to rapidly deteriorate. The part of the tube nearest the fire-box is the most effective heating surface; and the value of the heating surface of the tube rapidly decreases towards the smoke-box end.

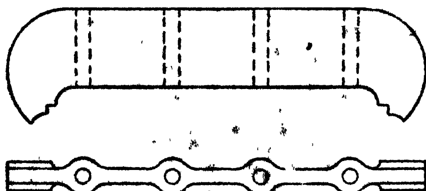


FIG. 133

The upper surface of the tube is also far more effective than the lower, even when the tube is clean; but when soot is deposited in the lower portion of the tube, that part of it is valueless as heating surface.

The chamber beyond the tubes and below the chimney is called the smoke-box, S B. A dome, S D, is usually provided,

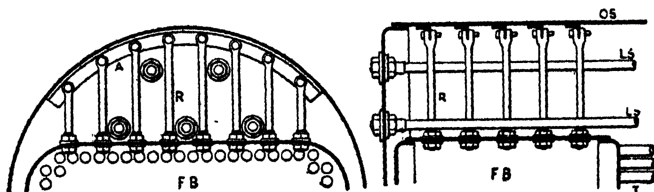


FIG. 134.

from which the steam is taken to supply the engines; and a safety valve, S V, is placed as shown.

Heating surface of tubes.—The student will be aware that, in order to obtain large heating surface in a boiler, a number of small tubes are used in preference to a few large ones. For the smaller the diameter of the tubes used to fill a given sectional area, the greater the area of heating surface obtained.

Thus, take a circle 1 in. in diameter, fig. 135, then its circumference$=1\times3{\cdot}1416$ ins. But if two circles, $\frac{1}{2}$ in. in diameter, be placed on the diameter of the 1-in. circle, touching each other and the large circle, then their circumferences $=(\frac{1}{2}\times3{\cdot}1416)\cdot2=3{\cdot}1416$ ins., the same as before; or, if 10 small circles, each $\frac{1}{10}$ in. diameter, be ranged along the same diameter, the sum of their diameters being 1 in., the sum of their circumferences is $(\frac{1}{10}\times3{\cdot}1416)\ 10=3{\cdot}1416$ ins. as before.

But, the smaller the circles used, the more room remains for the insertion of other circles within the area of the large circle; and, therefore, the smaller the diameter of the tubes, the greater the number possible in a given area, and the greater the heating surface obtained.

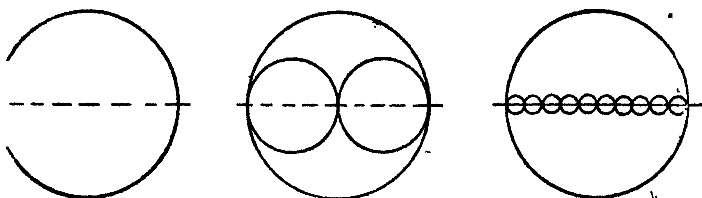


Fig. 135.

The practical limit to the diameter of the tube depends upon the possibility of keeping them from being choked up with soot and dirt. The tubes used in locomotive boilers are about $2\frac{1}{8}$ ins., and in marine boilers from $2\frac{1}{2}$ to $3\frac{1}{2}$ ins. outside diameter.

The student will note that the heating surface is measured from the *outside* diameter of the tube.

Safety Valves

The safety valve provides for the safety of boilers by allowing the steam to escape when its pressure exceeds a certain limit. The safety valve is kept in its place on its seating either by a weight at the end of a lever, by a strong spring, or by a heavy weight, placed directly over the valve, and these three forms will here be described.

A good safety valve is one which will not permit the pressure

in the boiler to rise above a fixed point, and, having reached that point, will allow all excess of steam to escape as fast as it is generated by the boiler.

Mr. Webb, of the London and North-Western Railway, in an experiment on a locomotive boiler fired hard, found that a pipe $1\frac{1}{4}$ in. diameter was sufficient to allow all the steam to escape as fast as generated without the pressure increasing beyond the initial pressure.

The Lever safety valve.—This valve (fig. 136) rests on a circular brass seating, and is prevented from rising by the steam pressure underneath the valve, by the weight at the end

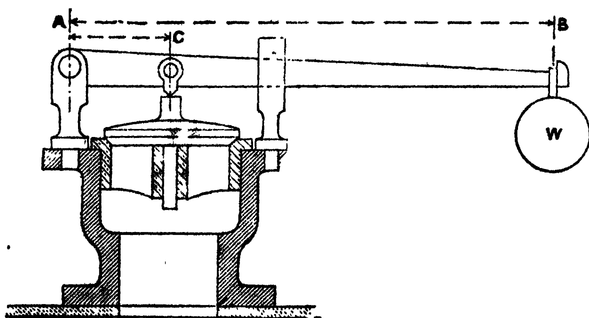


FIG. 136.

of the lever. The disadvantage of this valve is that it admits of being tampered with, and the effect of a small addition to the weight is magnified considerably in its action on the valve.

To find the weight W, or length of lever A B, for a given pressure of steam:

Let A B = length of lever from fulcrum A to centre of weight W.

A C = distance between centre of valve and fulcrum.

W = weight at end of lever.

w = weight of lever acting at centre of gravity of lever, assumed at centre of lever.

P = pressure of steam per sq. in.

a = area of valve.

V = weight of valve.

(1) If the effect of the weights of valve and lever be omitted, we have, when valve is just about to lift,—

Downward pressures = upward pressures.

Pressure on valve due to W.		Upward steam pressure.

$$W \times \frac{AB}{AC} = Pa$$

$$W = Pa \times \frac{AC}{AB}$$

(2) Taking the effect of weights of valve and lever into account (which should always be done where accuracy is required) we have, when valve is about to lift,—

Downward pressures = upward pressures.

Pressure on valve due to W.	Pressure due to weight of lever.	Weight of valve.	Total upward pressure on valve.

$$W \times \frac{AB}{AC} + w \times \frac{AB}{2AC} + V = Pa$$

Example.—Let it be required to find weight W at end of lever when $AB = 36$ ins., $AC = 4\frac{1}{2}$ ins., $w = 5$ lbs., $V = 2$ lbs., $P = 100$ lbs., and $a = 5$ sq. ins.

From (1) omitting weight of valve and lever we have

$$W = (100 \times 5) \times 4\tfrac{1}{2} \div 36$$
$$W = 62\tfrac{1}{2} \text{ lbs.}$$

From (2) including weight of valve and lever

$$\left(W \times \frac{36}{4}\right) + \left(5 \times \frac{36}{2 \times 4\frac{1}{2}}\right) + 2 = 100 \times 5$$
$$8W + 20 + 2 = 500$$
$$W = 59\tfrac{3}{4} \text{ lbs.}$$

These results show that, if a weight of $62\frac{1}{2}$ lbs. was placed on the lever instead of the proper weight, $59\frac{3}{4}$ lbs., the valve would not blow off at 100 lbs. as required.

A suitable length of lever AB for a given weight W may be obtained from the same equations.

At least two valves are fitted to each boiler. The valve seating may be either flat or coned to 45°. A bearing surface on the edge of the seating, $\frac{1}{16}$ in. in width, is found to be quite sufficient, and to answer better than a wider surface.

Spring-loaded Safety Valve for Locomotive

The following diagram, fig. 137, illustrates what is known as Ramsbottom's safety valve. It consists of two separate valves and seatings A A, having one lever, B, bearing on the two valves, and loaded by a spring D, the spring being placed between the valves. The tension on the spring can be adjusted by the nut E. By pulling or raising the lever B, the driver can relieve the pressure from either valve separately, and ascertain that it is not sticking on the seating.

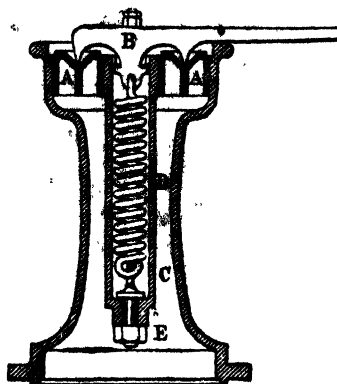


Fig. 137.

The Dead-weight Safety Valve

Fig. 138 illustrates a dead-weight safety valve as used for stationary boilers. The valve *a* rests on the seating *b*, which is fixed on the top of a long pipe, as shown. The valve is secured to a large casting A, which fits down over the pipe like a cap. This casting is provided with a ledge on which circular rings of metal, which act as weights, may rest.

To find the dead weight required (including casting and weights) for a valve of given area: Multiply the area of the valve by the pressure per square inch at which the valve is required to lift. Thus a valve 3 ins. diameter

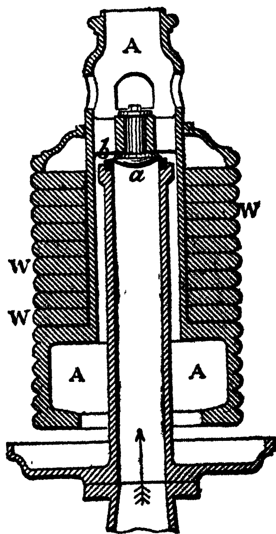


Fig. 138.

to blow off at 100 lbs. pressure requires the following dead weights :

$$\text{area} \times \text{pressure per sq. in.} = 3 \times 3 \times \cdot 7854 \times 100$$
$$= 706 \cdot 86 \text{ lbs. dead weight.}$$

Steam Regulating Valves

The *stop valve* is used to open or close the communication between the boiler and engine. A common form of valve is

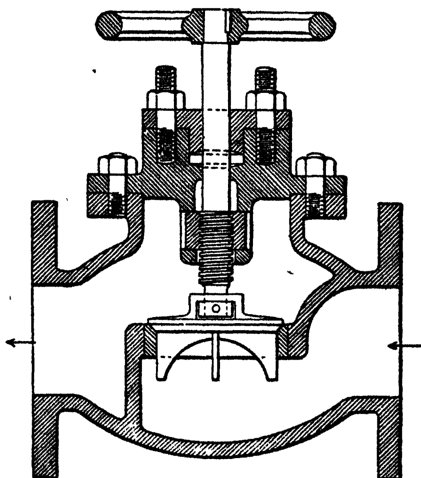


Fig. 139.

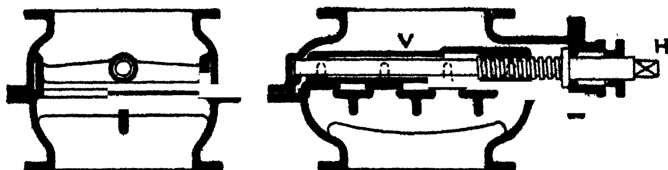


Fig. 140.

shown in fig. 139. It consists of a valve which may be opened or closed by means of a screwed spindle which is turned by a hand wheel.

The *gridiron valve* (fig. 140) is an arrangement for giving a large opening to the passage of steam with a comparatively small travel of the valve. It consists of a flat valve composed of a number of bars which move on a seating, having a number of ports or openings which are covered by the valve, as shown in the figure.

The valve V is opened or closed by the screw turned by a handle at H.

The equilibrium double-beat valve (fig. 141) consists of two disc-valves, A and B, on one spindle, each of which has its own seating. The arrows show the direction of the steam on entering the valve box from the passage E. The valve B is made a little larger than A, to enable the valve A to be put in its place from the top. The pressure acts on the top of one valve and on the bottom of the other, hence the two valves are nearly in equilibrium and may be easily lifted from their seating when under pressure.

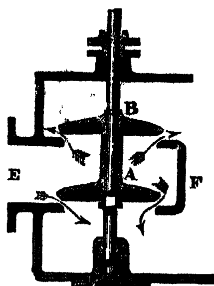


FIG. 141.

This arrangement provides a large opening to steam with valves of comparatively small diameter.

BOURDON'S PRESSURE GAUGE

The pressure gauge registers the pressure of steam in the boiler above the pressure of the atmosphere. The following figure (fig. 142) illustrates the construction of Bourdon's gauge, which is the one commonly used.

The gauge consists of a curved tube, B B, of a flattened or elliptical cross-section, as shown enlarged at C. The tube is closed at one end and open at the other, by which the interior of the tube is put in communication with the boiler pressure through the cock A. The closed end of the tube is attached (as shown in the figure) to a sector D, provided with teeth which gear with those of a small pinion on the same axis as the finger E F on the face. The effect of steam pressure in the

curved tube is that the tube *tends to straighten itself*, and thus, as the pressure increases, the closed end moves the sector, which acts on the finger and indicates the pressure. These gauges are carefully graduated by comparing their indications with those of a mercurial gauge.

The *efficiency of the boiler* is a fraction, and is estimated thus:

$$\text{Efficiency} = \frac{\text{water evaporated per lb. of fuel}}{\text{theoretical evaporative power of fuel}}.$$

Thus a good stationary boiler evaporates 10 lbs. of water per lb.

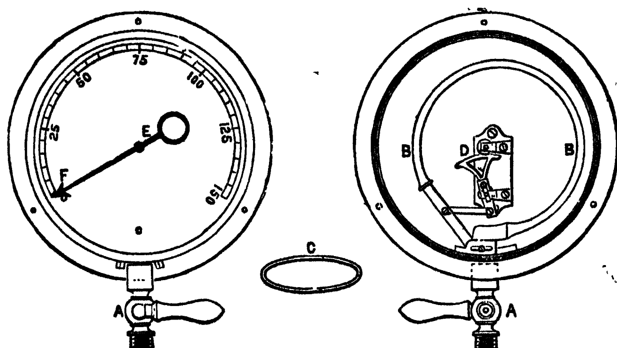


FIG. 142.

of fuel; but the theoretical evaporative power of the fuel is estimated at 14 lbs.

$$\text{Therefore efficiency of boiler} = \frac{10}{14} = \cdot 714, \text{ or } 71 \cdot 4 \text{ per cent.}$$

The pounds of water evaporated per pound of fuel varies with different types of boilers, as will be seen from the following approximate results:

	lbs. of water evaporated per lb. of coal.
Lancashire boilers, with water tubes .	11
Locomotive boilers	10
Marine boilers	8

	lbs. of water evaporated per lb. of coal.
Cornish boilers	8
Torpedo-boat locomotive boiler . .	7·5

From the above we see that the evaporative efficiency of the Lancashire water-tube boiler is high, while that of the torpedo-boat locomotive type of boiler is low. But it will be noticed that nothing is here said as to the *time* taken—which, in practice, is a highly important point. It is more important, for example, that a torpedo-boat boiler should be small, and yet capable of rapidly generating large quantities of steam, than that it should be economical. It is largely so also with the locomotive boiler. Such boilers, therefore, require

(1) to be strong enough to carry steam at the highest pressures;
(2) to have large and efficient heating surface;
(3) to maintain a vigorous combustion.

The strongest form of boiler is the cylindrical form of small diameter. The requisite heating surface of the locomotive type of boiler is provided by the large number of tubes of small diameter; and the vigorous combustion is obtained by the exhaust steam (or 'blast') passing into the chimney, which acts as a pump, creating a vacuum in the chimney and drawing the air through the fire-bars, thus providing a very strong draught, causing intense combustion in the furnace, and at the same time drawing the flame and hot gases through the tubes, and bringing them into contact with the large heating surface here provided.

The same effect is produced in torpedo-boat boilers, and in modern man-of-war boilers, by means of forced draught supplied by a fan, worked by the main machinery or by a separate small engine. In this way the combustion is much more perfect, the temperature of the furnace is greatly increased, and the escape of unburnt gas and smoke prevented; the wear and tear, however, is greatly increased.

The efficiency of boilers with respect to the *rate of evaporation* may be seen from the following table, taken from Hutton's 'Practical Engineer's Handbook':

Description of boiler.	lbs. of water evaporated per sq. ft. of heating surface per hour.
Vertical boiler, with cross tubes	2·20 lbs.
Vertical tubular boiler	2·25 „
Cornish boiler	2·30 „
Lancashire boiler	2·50 „
Marine boiler	5·00 „
„ with forced draught	13 to 15 lbs
Locomotive boilers	8 to 9 lbs.
Torpedo-boat boilers, locomotive type (forced draught)	18 lbs.

CHAPTER XVIII

PRACTICAL NOTES ON THE CARE AND MANAGEMENT OF ENGINES AND BOILERS

(1) BEFORE getting up steam the boiler water-gauge cocks should be tried to see that the water is in the boiler.

(2) The stop-valve should be opened a little, before the fire is lighted, so that, while the steam is being generated in the boiler, it may pass through the cylinders and jackets and warm them *gradually*, the temperature rising as the pressure rises. Meanwhile all drain-cocks from the slide jackets and cylinders should be opened to allow the steam to flow through, and the condensed steam to pass away. This will prevent the possibility of the cylinder cracking owing to sudden admission of hot steam against the cold metallic walls of the cylinder. This is especially important in cold weather.

(3) The drain-cocks should remain open for a few revolutions till all water has been blown out of the cylinder, and then closed.

(4) Should these precautions not have been attended to, then, since the exhaust port closes before the end of the stroke, the water in the cylinder would be compressed, and a difficulty found in starting the engines. Any attempt to force the engine by 'barring round' would tend to burst the cylinder cover, or to push the slide-valve off the face of the ports.

(5) See that all the lubricators are in good condition, the holes clear, and the worsteds clean, and that the lubricators are well supplied with oil.

(6) Should there be any tendency to heating of the bearings, the cap nuts should be eased and the lubricator examined to

see whether it is working properly. Should the bearing be very hot, the engine must be stopped, the cap removed, and the brass taken out and examined to see the cause.

(7) If a condensing engine, the vacuum gauge should be watched; and, if the vacuum is not maintained, the injection, or circulating water, should be regulated. If this does not produce the desired effect, there is probably an *air leak* through the piston-rod gland, or the air-pump-rod gland, which should be screwed up; and, if the vacuum is still defective, the cause must be looked for in the foot and head valves or the air-pump bucket valve (if any), or in leaky condenser tubes.

(8) See that the water in the boiler-gauge glass is kept at the proper height, namely, about half-way up the glass, and that the fires are kept in proper condition, and that the steam pressure is kept uniform. The feed water supply should be as uniform as possible, and not be shut off at one time, and wide open at another.

(9) When feeding the furnace the coals should be laid on in thin layers, and in small quantities at a time, care being taken to fill up all hollow places, and to keep the fire level. The fire-door should not be kept open a moment longer than is necessary.

(10) The damper regulating the draught should be kept only sufficiently open to generate the quantity of steam required.

(11) The ashes should not be allowed to accumulate in the ashpit, because the heat from them may cause the fire-bars to bend under the weight of the fuel in the furnace.

(12) To clean the fire, which should be done when it is dirty from the presence of clinker, scrape the fire from one side of the furnace to the other with the slice-bar, then break up the clinkers from the fire-bars with the slice-bar and draw them out with the rake with as much speed as possible. As soon as this half of the furnace is clear of clinker, turn the fire over from the other side on to the clean side, and throw a little round coals on the fire before cleaning the second half; then clear of clinker as before. Now level the fire over the bars with the slice-bar, and throw on a thin layer of round coals, and close the fire-door.

(13) All cocks and valves connected with the boiler should be moved daily, especially the safety valve. In the Navy the safety valves are lifted at least once every watch, to see that they are in working order.

(14) Should the engines stand idle for any length of time they should be turned partly round each day.

To test for a leaky slide valve.—Block the fly-wheel when the slide valve is in the middle of its stroke (seen by the position of the eccentric, which is in mid position a little before the piston reaches the end of the stroke) and open the indicator taps, or the relief cocks, or look at the exhaust pipe. A steady escape of steam indicates a leaky valve.

To test for a leaky piston.—Block the fly-wheel when the piston is situated at a short distance beyond the beginning of the stroke. Admit steam to the piston and open the indicator tap, or relief cock, on the exhaust side of the piston. An escape of steam will indicate a leaky piston. The leak may be caused by a leaky slide valve, so this should be tested first.

Annual Inspection of Engines and Boilers

Engines.—(1) Take off slide-valve cover and examine valve faces and fastenings of valves to rods; see that surfaces are all clean and bright; remove fatty substances which have accumulated from lubrication. Turn engines round and test lead of valve.

(2) Take off cylinder cover, examine cylinder for cracks or other defects.

(3) Examine condition of piston whether steam-tight, and its attachment to piston rod; take off junk ring, remove springs and spring rings, and see that they are in good condition.

(4) Air-pump and condenser to be opened out and cleaned, foot and head valves, bucket valves, and bucket packing to be examined, and defects made good. Also see that the injection valve and orifice is in good condition, and that air-pump rod is properly secured to bucket. If surface condenser, tube packings renewed where necessary.

(5) Caps to be taken off main bearings, cap brasses taken out and adjusted, and oil-ways cleaned.

(6) Connecting-rod brasses to be examined and adjusted if necessary.

To adjust connecting-rod brasses.—When fitted with liners or distance pieces between the two half brasses, remove the liners, screw down the brass on the journal and measure the distance between the two brasses with a pair of internal callipers. Take a gauge from this with external callipers and fit the liner tight between the brasses. Then slack the nuts off, put the liners in their places, and screw up the nuts. The brasses will then fit properly on the journal.

When the brasses are not fitted with liners, place a piece of thin lead wire on the journal and tighten up the brasses upon it. When the brasses are right-up, the lead wire will be flattened, and the thickness of the flattened wire will indicate the amount the brasses require to be set up. The proper freedom of the journal in the brass may be tested by disconnecting the other end of the rod, and swinging the rod on the journal; when tightened up, the rod should move freely on its bearing without any tendency to grip the journal.

(7) All stuffing boxes to be repacked.

Boilers.—(1) Stationary boilers, even when using clean feed water, should be opened and thoroughly cleaned out at least once a year, and all parts of the boiler and the boiler fittings carefully examined. The frequency of cleaning out the boiler will depend upon the kind of service and the character of the feed water: for example, locomotive boilers are cleaned out two or three times a week.

(2) Find the water-line inside the boiler and trace with a paying hammer for defective parts. Examine carefully for indications of pitting &c. Where decay has commenced, it should be carefully watched and steps taken to prevent further corrosion by scraping off all rust, and coating with a thin wash of Portland cement, or other substitute. Test the thickness of the plate where suspiciously thin by drilling a hole; tap it, and put in a screwed plug. Test the boiler by hydraulic pressure to twice its working pressure.

(3) Safety valves.—Remove the weights or springs from the valve, and take the valve out to clean and examine it; see that the seating is not pitted and that the valve works freely. Weigh the weights on the valve (or test the springs), and divide the sum by the area of the valve. This will give the pressure per sq. in. at which the valve will lift. Check this result by the pressure gauge.

(4) Stop-valve.—Remove the cover and take out and examine the valve and its seating.

(5) Water-gauge cocks.—Take all plugs out and see that all the passages are clear.

(6) Feed-valve.—Take out and examine the condition of valve and its seating.

(7) Blow-off and scum cocks.—Take out, clean, and examine condition of plugs and bearing surfaces. Also examine the gland bolts of these cocks; if they break, the plug is blown out.

(8) Fire-bars.—See that the fire-bars are not too far apart. To fit fire-bars, fill the furnace tight with bars, then remove one bar; this will allow for expansion.

APPENDIX

MENSURATION OF SURFACES AND VOLUMES

1. Area of rectangle . . = length × breadth.
2. Area of triangle . . = base × $\frac{1}{2}$ perpendicular height.
3. Diameter of circle . . = radius × 2.
4. Circumference of circle . = diameter × 3·1416
5. Area of circle . . . = diam. × diam. × ·7854
6. Area of sector of circle . = $\frac{\text{area of circle} \times \text{No. of degrees in arc}}{360}$.
7. Area of surface of cylinder = circumference × length + area of two ends.

8. To find diameter of circle having given area: Divide the area by ·7854, and extract the square root.

9. To find the volume of a cylinder: Multiply the area of the section in square inches by the length in inches = the volume in cubic inches. Cubic inches divided by 1728 = volume in cubic feet.

QUESTIONS AND EXERCISES

I

1. Explain the nature of the phenomenon which we call 'heat.'

2. Distinguish between 'temperature' and 'quantity of heat.'

3. Convert 5°, 14°, 41°, 68°, 158°, 266° Fahrenheit to Centigrade; and 1°, −30°, −25°, 90°, 120° Centigrade to Fahrenheit.

4. A Fahrenheit thermometer rises through 45°; how many degrees would this rise indicate on the Centigrade thermometer?

5. What is meant by the 'specific heat' of a substance? One ounce

of copper at 212° F. is immersed in 1 lb. of water 55° F.; find the increase of temperature of the water.

6. Express the following temperatures in degrees absolute: 60° F., 100° F., 247° F., and 0° C., 100° C.

7. Convert 338° F. and 165° C. into degrees Réaumur.

II

1. Define the 'unit of heat,' the 'unit of work,' 'horse-power,' and the 'mechanical equivalent of heat.'

2. Find the units of heat required to raise water from 55° F. to 212° F., and express the same in units of work.

3. One pound of water is heated from 60° F. to 100° F.; find the units of heat absorbed by the water, and the equivalent units of work.

4. Five-and-a-half pounds of water are heated from 50° F. to 75° F.; find the units of heat absorbed, and the equivalent units of work.

5. A weight of a ton is lifted by a steam engine to a height of 400 ft.; what amount of heat is consumed in the act?

6. Describe the experiment conducted by Joule to determine the mechanical equivalent of heat.

III

1. Give examples of good and bad conductors of heat.

2. Explain the process of heating water by 'convection.'

IV

1. Explain the process of the combustion of coal.

2. What conditions are necessary for the complete combustion of the gases distilled from the coal?

3. What is the effect of an incomplete supply of oxygen during the combustion of solid fuel?

4. Given that 966 units of heat are required to evaporate 1 lb. of water from and at 212°, how many lbs. of water should 1 lb. of coal evaporate whose total heat of combustion is 14,000 units per lb.?

5. What is the difference in the heat of combustion between carbon burnt to carbonic oxide, and carbon burnt to carbonic acid gas?

6. What horse-power should be obtained by burning 1 lb. of coal if there were no waste? (Sc. & A. 1883.)

7. An engine of 6,000 H.P. burns $1\frac{3}{4}$ lb. of coal per I.H.P. per hour; find the consumption of coal per 24 hours.

V

1. Give what practical illustrations you can of the expansion of metals by heat.

2. What is 'the law of Charles'?

3. A gas occupies 5·5 cub. ft. at 32° F.; what volume will it occupy at 212° F. under constant pressure?

4. A volume of air at 350° F. exerts a pressure of 60 lbs.; find its pressure when the temperature is reduced to 32° F.

5. A gas occupies 15·5 cub. ft. at 100° C.; what volume will it occupy at 0° C., the pressure on the gas remaining the same?

6. What do you understand by the term *absolute* pressure?

7. Explain the phenomenon of *boiling*.

8. What effect has pressure on the temperature at which water evaporates?

9. Describe a simple process of obtaining pure water from muddy water.

10. What is the meaning of the word 'vacuum'? and explain why it is impossible in practice to obtain a *perfect* vacuum.

11. Describe an experiment illustrating the reality of atmospheric pressure.

VI

1. Describe carefully the stages involved in the conversion of water into steam under a movable piston, noticing especially the changes in temperature and volume.

2. What work is done in raising a piston 2 sq. ft. in area through a height of 5 ft. against atmospheric pressure? and represent this by an area.

3. What is the total number of units of heat required to convert water at 32° F. into steam at 212°? and explain how these units have been expended.

4. In what way is the 'latent' heat expended during the formation of steam?

5. How is the *efficiency* of an engine expressed?

6. Considering the work done per lb. by steam during formation at varying pressures without expansion, what advantage is gained by using high pressures rather than low?

7. Given that the volume per lb. of steam at 120° is 3·65 cub. ft.; find the external work done per lb. during formation.

8. Find the weight of steam required per horse-power per hour in example 7.

9. What conditions affect the total quantity of heat rejected by steam to the condenser?

10. Define 'latent heat of steam' and 'total heat of evaporation.'

11. Give formulæ for finding the total and latent heats of steam, and apply them, having given that the temperature of steam at 115 lbs. is 338° F.

12. A cylinder is 16 ins. diameter, and stroke of piston 2 ft.; find the

area of the piston and the volume displaced by the piston at each stroke neglecting clearance.

13. If, in the previous question, the area and volume of the piston rod, $2\frac{1}{4}$ ins. diameter, be deducted, what will then be the effective area of the piston and volume of steam on the rod side?

14. The high-pressure cylinder of an engine is 36 ins. diameter, the initial pressure of steam 120 lbs. per sq. in.; find the load on the piston in tons.

15. The area of a piston is 706·8 sq. ins.; find the diameter of the air pump which is one-half that of the cylinder.

16. The cylinder of an engine is 74 ins. in diameter and the stroke is $7\frac{1}{2}$ ft.; what is the capacity of the cylinder? How many lbs. of water must be evaporated in order to fill such a cylinder with steam at an actual pressure of 15 lbs., it being given that steam at 15 lbs. pressure occupies a space equal to 1,670 times that of the water from which it is generated?

(Sc. & A. 1871.)

(*Note.*—1 lb. of water = ·016 cub. ft.)

VII

1. What is saturated steam?

2. What is the temperature of saturated steam at atmospheric pressure, also at pressures of 20, 60, 100, 150, 200 and 400 lbs. per sq. in.?

3. Show the relation between temperatures and pressures by a curve.

4. Draw a curve illustrating the relation between the volume and pressure of saturated steam.

5. If 2 lbs. of water at 200° F. be mixed with 2·5 lbs. of water at 212° F., find the temperature of the mixture.

6. How much water at 60° F. must be mixed with 1 lb. of water at 212° F., so that the resulting temperature may be 120° F.?

7. How much water at 60° F. will be necessary to condense 1 lb. of steam at 212°, so that the resulting temperature shall be 120° F.?

8. Find the temperature of the mixture when 17·63 lbs. of condensing water at 60° F. are used per lb. of steam at 212°.

VIII

1. What is Boyle's Law?

2. Steam is admitted into the cylinder of an engine at the pressure of 45 lbs. per sq. in. absolute, and is cut off at one-third of the stroke; find the pressure of the steam in pounds per sq. in. at half-stroke, and also at the end of the stroke.

3. Suppose the steam pressure in above example is 45 lbs. above the atmosphere, and the engine is non-condensing, what is the effective pressure on the piston at half-stroke and also at the end of the stroke?

4. Find the steam pressure at the end of the stroke of the piston in an

engine where the steam is admitted at a pressure of 30 lbs. above the atmosphere, and is cut off at two-fifths of the stroke. (Sc. & A. 1882.)

5. A steam cylinder is 4 ft. long; the steam enters at 60 lbs. boiler pressure and is cut off at one-third of the stroke; what is the steam pressure when the piston has travelled over 2 ft., 3 ft., and 4 ft. respectively? Give your answer in pressures above the atmosphere.

6. Steam is admitted into a cylinder at a pressure of 25 lbs. on the square inch above the atmospheric pressure of 15 lbs. on the square inch, and is cut off at such a point that its pressure at the end of the stroke is 5 lbs. below that of the atmosphere. At what point of stroke was it cut off? Make a diagram, showing approximately the steam pressure on the piston throughout the stroke. (Sc. & A. E. 1885.)

7. Draw the theoretical indicator diagram when steam at 75 lbs. boiler pressure is cut off and expanded to three times its initial volume, first by calculation, and then by the graphical method.

8. In a cylinder having a piston with a 4-ft. stroke, steam at 75 lbs. absolute pressure is cut off at two-fifths of the stroke; find the pressure of the steam at the second, third, and fourth foot of the stroke (neglecting the effect of clearance).

IX

1. Explain the use of the hyperbolic logarithm in obtaining the area of the theoretical indicator diagram.

Write down the expression for finding the total area of the figure.

2. Explain the reason of the advantage gained by using steam expansively, and compare the effect in work done, and in steam and fuel consumption with steam at 60 lbs. absolute pressure—

(*a*) when cut off at one-third of the stroke;

(*b*) when admitted throughout the whole stroke, omitting the effect of back pressure.

3. What is the effect of using a condenser on the total work done by the engine?

4. Write down the operation of finding the mean effective pressure represented by an indicator figure.

5. Steam is admitted to a cylinder at 75 lbs. absolute and cut off at one-third of the stroke; back pressure 15 lbs. Draw the theoretical indicator diagram, and find the mean effective pressure by measurement.

6. Write down the expression for finding the mean pressure by the use of a table of hyperbolic logarithms.

7. Given that the hyperbolic logarithm of 3 is 1·098; find the mean pressure in question 5.

8. Write down the formula for finding the indicated horse-power of an engine.

9. Find the indicated horse-power of an engine with a cylinder 16 ins.

diameter, length of stroke 2 ft., number of revolutions 70, mean effective pressure on piston 30 lbs. per sq. in.

10. A single-cylinder engine 24 ins. diameter, 3 ft. stroke, mean effective pressure of steam 40 lbs., makes 60 revolutions per minute; find its indicated horse-power.

11. The above 24-in. cylinder engine is required to indicate 250 horse-power; what must be the mean effective pressure of steam when running at the same speed?

12. Suppose that, in the previous case, instead of obtaining the 250 I. H. P. by increasing the mean pressure, it was done by increasing the number of revolutions; how many revolutions per minute must the engine now make?

13. Find the horse-power of a locomotive engine which can draw a train weighing 100 tons (including its own weight) along a level road at 30 miles per hour, the train resistance being taken at 10 lbs. per ton of load. (Sc. & A. 1884.)

14. What diameter of cylinder will develop 50 horse-power with a 4-ft. stroke 40 revolutions per minute, and a mean effective steam pressure of 30 lbs. above the atmosphere, the engine being non-condensing? (Sc. & A. 1883.)

15. In a beam engine the mean pressure of the steam on the piston is 20 tons, and the length of the crank is $2\frac{1}{2}$ ft.; what is the horse-power when the crank shaft makes 30 revolutions per minute? (Sc. & A. 1883.)

16. An engine is required to indicate 50 horse-power, with a mean effective pressure on piston of 35 lbs. per sq. in.; length of stroke 2 ft.; number of revolutions 60; find the diameter of the cylinder.

17. Compare the economical effect of using steam at 80 lbs. absolute, and steam at 40 lbs. absolute in a single-cylinder condensing engine. Back pressure 3 lbs., and terminal pressure 10 lbs. in each case.

18. Suppose steam at 60 lbs. boiler pressure is used to drive two engines, one a non-condensing engine, and the other a condensing engine. In the non-condensing engine the steam is cut off at $\frac{1}{3}$ of the stroke, back pressure 18 lbs. absolute; and in the condensing engine at $\frac{1}{6}$ of the stroke, back pressure 3 lbs. absolute. The cylinders of both engines are the same size. Draw the theoretical indicator diagrams and compare the relative work done, and weight of steam used in the two cases.

19. Explain in what way the amount of back pressure limits the number of useful expansions of the steam in the cylinder.

20. What is 'clearance'? and explain its effect on the work done by the steam in the cylinder.

21. How may the loss by clearance be modified?

22. What is 'priming'?

23. Explain in what way 'cylinder condensation' limits the useful range of expansion of the steam on the cylinder.

24. What are the remedies adopted to reduce the amount of condensation of the steam in the cylinder?

X

1. Make a hand-sketch of the cylinder (fig. 38).

2. How is the piston-rod made steam-tight in passing through the cylinder cover?

3. What are 'wire-drawing' and 'cushioning'?

4. Make a sketch of a cylinder fitted with a liner (see fig. 105).

5. What is the object of the cylinder escape valve, and relief cocks?

6. A cylinder is 36 ins. diameter, stroke of piston 3 ft. 6 ins.; find the capacity of the cylinder, allowing 7 per cent. in addition for clearance space.

7. Find the weight of steam occupying 26·47 cub. ft. at a pressure of 12 lbs. per sq. in. absolute; given that steam at 12 lbs. absolute occupies 31·9 cub. ft. per lb.

8. The weight of steam passing through the engine per stroke is ·83 lbs.; find the weight used per hour when the engine makes 85 revolutions per minute.

9. Sketch some form of steam-engine piston, and explain how it is made to work steam-tight in the cylinder.

10. What is the 'packing ring,' the 'tongue piece,' and the 'junk ring,' and what is the purpose of each?

11. What is the speed of the piston of a locomotive engine having 24 ins. stroke, with 7-ft. driving wheels when running at 40 miles per hour? (Sc. & A. 1880.)

12. What is the piston displacement per minute in an engine with 20 ins. diameter cylinder, 2 ft. 6 in. stroke, running at 60 revolutions?

13. What is the nature of the stress on the screwed end of the piston-rod during the to-and-fro motion of the piston?

14. Sketch some form of engine crosshead.

15. Explain the nature of the thrust on the guides.

16. Of what use is the top guide when the thrust is usually carried on the bottom guide?

17. Sketch some form of engine connecting-rod.

18. What are the 'dead centres'?

19. Draw a diagram showing the relative positions of piston and crank pin when the crank makes angles of 0°, 30°, 60°, 90°, 120°, 150°, 180°, when the length of the connecting rod is $1\frac{1}{2}$ times the stroke of the piston. Mark also upon the piston path, the piston positions for the same crank angles if the connecting rod were infinitely long.

20. In an engine with a cylinder 24 ins. diameter and 3 ft. stroke, the mean pressure of the steam on the piston is 45 lbs. per sq. in.; find the mean pressure on the crank pin in the direction of its motion.

21. The crank of an engine is 2 ft. long and the mean tangential force acting upon it is 17,000 lbs. What is the mean pressure of the steam upon the piston of the engine during each stroke? (Sc. & A. 1876.)

XI

1. Sketch a sectional view of the steam and exhaust ports of an engine showing a valve, without lap, at the end of its stroke. Show by arrows the direction of the steam.

2. Define 'outside lap,' 'inside lap,' and 'lead' of a slide valve; make sketches illustrating your answer.

3. The width of a steam port is $1\frac{1}{4}$ in.; the lap of the valve $\frac{9}{16}$ in., and the lead $\frac{1}{8}$ in. Draw a diagram giving the travel of valve and angular advance of the eccentric.

4. What is the effect of outside and inside lap of the valve?

5. Describe how you would set a slide valve.

6. Find the travel of a valve having $\frac{3}{4}$ in. outside lap, and maximum port opening $1\frac{3}{8}$ in.

7. Sketch a slide valve in mid position to the following dimensions: exhaust port 3 ins. wide, bars 1 in. wide, steam ports 2 ins. wide, outside lap $1\frac{1}{2}$ in. Sketch also the same valve at the beginning of the piston stroke with $\frac{1}{2}$ in. lead. (Sc. & A. 1882.)

8. Make a hand-sketch of a piston valve, describe its construction, and state what are its advantages.

9. Make a sketch of a double-ported slide valve, and explain how the pressure of steam at the back of the valve is largely removed.

10. Make a hand-sketch of an eccentric and describe its construction.

11. What is the object of the link motion? and explain how that object is accomplished.

12. Make a sketch showing the approximately relative position of crank and eccentrics in a link motion.

XII

1. Sketch a locomotive crank axle.

2. What is meant by 'the tangential pressure on the crank pin'? and how may it be determined geometrically, assuming the pressure is uniform throughout the stroke?

3. What are the advantages of having two cranks at right angles rather than together or exactly opposite each other?

4. Sketch a pedestal suitable to carry a shaft when the resultant load on the bearing is inclined to the vertical.

5. Why is it important that the bearings of shafts should be made sufficiently large?

XIII

1. What is the object of the condenser?

2. Make a sketch of a jet condenser, and sketch and explain the means adopted for removing the water from the condenser.

3. Sketch and describe the surface condenser.

4. Explain the circumstances which led to the abandonment of the jet condenser in favour of the surface condenser in steamships.

5. Make a sketch showing how the tubes are secured in the condenser tube plate.

6. The index finger of a vacuum gauge points to 26. Explain the meaning of this.

7. Make a sketch of a feed pump, and explain its action.

8. The diameter of the plunger of a feed pump is 6 ins., length of stroke 10 ins.; find the capacity of the pump.

9. Find the weight of water thrown per minute by a pump 1,000 cub. ins. capacity, and 25 deliveries per minute.

10. A pump is 1 ft. 9 ins. diameter; length of stroke, 2 ft. 6 ins.; the bucket is covered with water at each stroke to a height of 2 ft.; revolutions of engine 50 per minute; find the weight of water lifted per hour.

11. A pump valve is made in the form of two rings, each 1 in. wide, and of internal diameter 5 and 10 ins. respectively; what is the area of the openings in the seating?

12. A pump valve is 3 ins. in diameter; what should be its lift so that the opening for escape of water shall be the same as if there were no valve?

13. A surface condenser has 1,725 tubes, each 13 ft. long, and $\frac{3}{4}$ in. outside diameter; what amount of condensing surface do they give? Write down two numbers which express pretty nearly the relative conducting powers of copper and iron. (Sc. & A. 1876.)

XIV

1. Sketch a Watt governor, and explain the object of the governor.

2. Say whether the governor fulfils its purpose perfectly, and give your reasons for your answer.

3. Describe the construction of the 'Porter' governor, and sketch an arrangement showing how it may be made to act upon an expansion valve.

4. What is the object of the fly-wheel?

5. Make a sectional sketch of a locomotive showing the arrangement of the engine underneath the boiler.

XV

1. What is the reason for the adoption of the compound engine, and in what respects is this engine superior to the single-cylinder engine?

2. Make a hand-sketch of the cylinders of the compound engine in figs. 105 and 106, and explain how the steam is supplied to and exhausted from each cylinder.

3. How is the low-pressure cylinder of a compound engine proportioned?

4. Find the number of expansions of the steam in a compound engine when the piston diameters are as 1 to 2, and the cut-off in the high-pressure cylinder is at half stroke.

5. Find the point of cut-off in the high-pressure cylinder of a two-cylinder compound condensing engine, when the volumes of the cylinders are as 1 to $3\frac{1}{2}$; initial pressure 90 lbs. by boiler gauge and terminal pressure 10 lbs. absolute; allowing a loss of 5 lbs. between boiler pressure and initial pressure in the cylinder.

XVI

1. Make a skeleton sketch of a compound tandem engine.

2. Draw a theoretical indicator diagram illustrating the distribution of the steam in the Woolf engine, assuming a cut-off at $\frac{1}{3}$rd of the stroke in the high-pressure cylinder, taking steam to end of stroke in low-pressure cylinder, and ratio of cylinder volumes 1 to 3. Combine the diagrams.

3. Compare, by an example, the range of temperatures in the separate cylinders of a compound engine, with the range on a single-cylinder, using the same number of expansions of the steam.

4. Show that the variation of stress on the mechanism of the engine is less in the compound engine than in the single-cylinder engine, working with the same weight of steam through the same range of pressures.

5. Make a skeleton sketch of a two-cylinder compound receiver engine.

6. Explain the distribution of the steam in the two-cylinder compound receiver engine, and illustrate your answer by drawing the theoretical diagram for a cut-off at half stroke in both cylinders.

7. Write what you know of the improvement in coal consumption which has taken place with the introduction of the various types of compound engines.

8. How do you account for this improvement?

1. Compare the resistance of cylindrical vessels to internal pressure, longitudinally and transversely.

2. Make a sketch showing how water tubes are fitted to boiler furnace flues.

3. Draw a longitudinal section of a Lancashire boiler, showing all the necessary fittings.

4. Sketch a longitudinal section of a marine boiler, and explain how the boiler is stayed.

5. A steamship has two boilers, each with three furnaces, 3 ft. diameter by 6 ft. long; find the fire-grate area.

6. Find the heating surface of a marine boiler of the following dimensions:

(*a*) 3 Furnaces, each 3 ft. diameter × 6 ft. long.
(*b*) 3 Combustion chambers:
Top plates 2 (3 ft. 6 ins. × 2 ft. 3 ins.)
1 (3 ft. 0 in. × 2 ft 3 ins.)
Back plates 2 (3 ft. 6 ins. × 3 ft. 6 ins.)
1 (3 ft. 0 in. × 5 ft. 0 in.)
Side plates 4 (2 ft. 3 ins. × 3 ft. 6 ins.)
2 (2 ft. 3 ins. × 5 ft. 0 in.)
(*c*) 3 Back tube plates:
2 (3 ft. 6 ins. × 3 ft. 0 in.)
1 (3 ft. 0 in. × 4 ft. 6 ins.)
Less area of 200 holes, 3 ins. diameter.
(*d*) 200 tubes, 3 ins. external diameter, length between tube plates 6 ft. 3 ins.

7. Make a sketch of the longitudinal section of the locomotive boiler, showing how the flat crown of the surface is stayed.

8. Sketch a method of staying the flat crowns of furnaces by bar stays and angle-iron.

9. Illustrate the advantage of small tubes over large ones, to provide large area of heating surface.

10. Make a sketch of a lever safety valve.

11. A valve, 3 ins. diameter, is held down by a lever and weight, length of the lever being 10 ins., and the valve spindle being 3 ins. from the fulcrum. You are to disregard the weight of the lever, and to find the pressure per square inch, which will lift the valve when the weight hung at the end of the lever is 25 lbs. (Sc. & A. 1881.)

12. Sketch a Ramsbottom safety valve.

13. Find the dead weight required for a valve $3\frac{1}{2}$ ins. diameter required to blow off at 90 lbs. per sq. in.

14. Sketch an equilibrium double beat valve.

15. Describe the construction and action of Bourdon's pressure gauge.

16. Find the efficiency of a boiler which evaporates 11 lbs. of water per lb. of fuel.

XVIII

1. What precautions would you take when getting up steam?

2. Suppose the vacuum in the condenser was not satisfactory, what would you do?

3. What points should be attended to by a man in charge of a boiler?

4. How would you test (*a*) for a leaky slide valve; (*b*) for a leaky piston?

5. How would you adjust the brasses to their journal, after the journal had worked loose by wear?

ANSWERS

I

(3) −15°, −10°, 5°, 20°, 70°, 130° C.; and 33·8°, −22°, −13°, 194°, 248° F.
(4) 25°. (5) 1° nearly. (6) 520°, 560°, 707°, and 273°, 373°.
(7) 136° R., and 132° R.

II

(2) 157 units of heat; or 121,204 units of work.
(3) 40 units of heat; 30,880 units of work.
(4) 137½ units of heat; 106,150 units of work. (5) 1160·6.

IV

(4) 14·5 lbs. (5) 10,100 units. (6) 5 H.P.
(7) 112·5 tons.

V

(3) 7·5 cub. ft. (4) 36·47 lbs. (5) 11·34 cub. ft.

VI

(2) 21,168 ft. lbs. (7) 63,072 ft. lbs. (8) 31·39.
(11) 1183·4 and 878·4. (12) 201 sq. ins.; 4,824 cub. ins.
(13) 197·024 sq. ins.; 4728·576 cub. ins. (14) 54·53 tons.
(15) 15 ins. (16) 224 cub. ft.; 8·38 lbs.

VII

(5) 206·66° F. (6) 1·53 lbs. (7) 17·63 lbs. (8) 120° F.

VIII

(2) 30 and 15. (3) 25 and 5. (4) 18 lbs. absolute.
(5) 35, 18·33, 10. (6) ¼. (8) 60, 40, 30.

IX

(5) 37·4. (7) 37·4. (9) 51·16. (10) 197·4.
(11) 50·66. (12) 76. (13) 80. (14) 14¾.
(15) 407·27. (16) 16 ins.

X

(6) 26·47 cub. ft. (7) ·83 lbs. (8) 8466 lbs.
(11) 640 ft. per min. (12) 654·16 cub. ft. per min.
(20) 26703·6 lbs.

XI

(6) $4\frac{1}{4}$ ins.

XIII

(8) 282·7 cub ins. (9) 904·2 lbs. (10) 900,000 lbs.
(11) 18·85 and 34·46 sq. ins. (12) $\frac{3}{4}$ inch.
(13) 4,403 sq. ft.; 6 to 1.

XV

(4) 8. (5) $\frac{7}{20}$.

XVII

(5) 108 sq. ft. (6) 1297·27 sq. ft. (11) 11·9 lbs. per sq. in.
(13) 865·89 lbs. (16) ·785.

SCIENCE AND ART DEPARTMENT EXAMINATION PAPER—1889

SUBJECT XXII. STEAM

First Stage or Elementary Examination

INSTRUCTIONS.

You are not permitted to attempt more than *six* questions.

The value attached to each question is shown in brackets after the question.

1. In an atmospheric pumping engine how is the injection water and condensed steam got rid of? Sketch and explain some of the principal improvements made by Watt in engines of this kind. (15.)
2. Describe, with sketches, the alterations made by Watt in order to convert a single-acting into a double-acting engine. (15.)
3. What is the latent heat of steam at 212° F. expressed in foot-pounds? If 1 lb. of steam at 212° F. is mixed with 10 lbs. of water at 60° F., find the resulting temperature. (15.)
4. Steam expands in the cylinder of an engine from a pressure of 30 lbs. above the atmosphere to 5 lbs. below the atmosphere, at what part of the stroke was the steam cut off? The pressure of the atmosphere may be taken at 15 lbs. (15.)

5. Describe, with sketches, the construction of a horizontal direct acting engine, working with high-pressure steam and without condensation, showing how the steam is admitted into the cylinder and let out again as required. (20.)
6. Define the lap of a slide valve, and explain your answer by reference to a sketch. Account for the difference in the working of two engines, one of which has lap on the steam side of this valve and the other has not. (15.)
7. Sketch the end of the connecting rod of a locomotive engine which embraces the crank pin. Show clearly the method of tightening the brasses on the crank pin by a gib and cotter. (15.)
8. Marine engines are fitted with a so-called air pump, circulating pump, feed pump, and bilge pump. What are the respective uses of these pumps? Show, with a sketch, the construction of any one of them, and explain how it acts. (15.)
9. The stroke of a direct acting engine is 5 feet, and the crank shaft makes 30 revolutions per minute, find the mean speed of the piston in feet per minute. State your reasons for concluding that there is no loss of work from the oblique action of the connecting rod during successive portions of the stroke. Friction is neglected. (15.)
10. Why are the longitudinal joints in cylindrical boilers usually double riveted, while the transverse joints are only single riveted? Sketch in longitudinal section a marine high-pressure boiler. (15.)
11. Sketch and describe the following safety valves :—
    (1) An ordinary lever valve.
    (2) The Ramsbottom spring loaded valve.
    (3) An ordinary dead weight valve. (20.)
12. Explain the principle of Watt's pendulum governor for a steam engine, and sketch the apparatus. Why is it important that the points of suspension of the arms should be near to the vertical axis of rotation? (15.)
13. Compare the crank with the eccentric. Show that they both produce the same motion. State reasons for employing one or the other in particular cases. (15.)
14. Sketch a longitudinal sectional elevation through the cylinder and condenser of a trunk engine, showing the air-pump and valves. How are the piston and trunk kept steam-tight? (20.)

INDEX

ABSOLUTE temperature, 6
Air pump, 110
— vessel, 118

BACK pressure, 53
Boilers, 158
— stationary, 160
— Lancashire, 161
— vertical, 165
— marine, 166
— locomotive, 169
— efficiency of, 176
Boiling, 23
— point, 3
Bourdon's gauge, 177
Boyle's Law, 42
Brasses, to adjust, 184

CAPACITY of pumps, 118
Centigrade thermometer, 3
Charles, law of, 21
Clearance, 63
Combustion, 15
— heat of, 17
— chamber, 166
Compound engines, 129
— types of, 139
Condensation, 25
Condensers, jet, 109
— surface, 112
— tubes, 114
Condensing water, 40
Conduction, 12
Connecting rod, 83
Convection, 13
Couplings, 107
Cranks, 101
Crossheads, 80
Curve, hyperbolic, 44
Cushioning, 68
Cylinder, details of, 72
— condensation, 65
— escape valve, 74
— relief cocks, 75

DEAD centres, 85
— plate, 161
Double-beat valve, 177
— ported valve, 96

ECCENTRIC, 97
Energy, 9
Engines, non-condensing, 69
— compound, 129
— locomotive, 125
— receiver, 146
— Woolf, 139
— management of, 181
Equilibrium valve, 177
Evaporation, rate of, 179
Evaporative power of fuel, 18
Expansion, economy of, 52
— limit of, 61
— of solids, 20
— of gases, 21
— of steam, 48

FAHRENHEIT thermometer, 3
Formation of steam, 28
Freezing point, 3
Fuel, evaporative power of, 18

GAUGE glass, 164
Gland, 73
Governors, 119
Grate area, 168
Guides, 80
Gusset stays, 164

HEAT, 1
— latent, 30
— mechanical equivalent, 10
— sensible, 36
— total, 36
— transfer of, 12
— unit of, 7
Heating surface, 168, 171
Horse-power, 9
— indicated, 57
Hyperbolic curve, 44

INDICATED horse-power, 57

JACKET, steam, 68, 74
Joule's experiment, 10
Journals, 108
Junk ring, 77

LANCASHIRE boiler, 161
Lap, effect of, 94
— inside and outside, 92
Law of Boyle, 42
— Charles, 21
Lead, 92
Leaky piston, to test, 183
— valve, — 183
Lever safety valve, 173
Link motion, 98
Locomotive, the, 125
— boiler, 169

MARINE boiler, 166
Mean pressure, 55
Mechanical equivalent of heat, 10
Mensuration, 186
Mixtures, temperature of, 40

PEDESTALS, 108
Pistons, 76
Piston displacement, 79
— rods, 80
— speed, 79
— leaky, 187
— valve, 95
Porter governor, 123
Pressure, absolute, 22
— of the air, 22
— gauge, 177
— mean, 55
Priming, 65
Pumps, 116
— capacity of, 118

QUANTITY of heat, 3
Quadruple expansion engines, 154
Questions, 186

RADIATION, 12
Range of temperature, 130, 144
Réaumur thermometer, 3
Receiver engines, 146
Reversing gear, 98
Rotary engines, 89

SAFETY valve, 172
— lever, 173
— spring, 175
— dead weight, 175
Saturated steam, 38
Shafts, crank, 101
Shaft couplings, 107
Shrinking on, 101
Slide valve, 89
— to set, 94
— double ported, 96
Specific heat, 5
Spring rings, 76
Steam, expansion of, 48
— formation of, 28

Steam, heat rejected by, 34
— properties of, 39
— regulating valve, 176
— saturated, 38
— weight of, 39, 75
— work done by, 29
— volume of, 39, 75
Stop valve, 176
Strap, gib, and cotter, 84
Stuffing box, 73

TABLE of specific heats, 5
— heat of combustion, 17
— steam properties, 39
Tangential pressure, 102
Temperature, 3
— absolute, 6
— of mixtures, 40
— range of, 130, 144
Thermometers, 3
— compared, 4
Triple expansion engines, 153
Tubes, boiler, 168
— condenser, 114
Turning effort, 102

UNIT of heat, 7
— of work, 8

VACUUM, 25
— gauge, 115
Valve, double-beat, 177
— double ported, 96
— gridiron, 177
— slide, 89
— stop, 176
— lap and lead of, 92
— lift of, 117
— piston, 95
— regulating, 176
Vertical boiler, 165
Volume of steam, 39, 75

WATER, condensing, 40
— weight of, 118
— tubes, 161
Watt governor, 119
Weight of steam, 39, 75
Wire-drawing, 73
Woolf engines, 139
Work, unit of, 8

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